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OF

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AND OF THE

INDICATOR AND PRONY BRAKE.

FOR ENGINEERS AND TECHNICAL SCHOOLS.

BY

R. H. THURSTON, M.A., LL.D., DR.ENG'G;
DIRECTOR OF SIBLEY COLLEGE, CORNELL UNIVERSITY;
PAST PRESIDENT AM. SOCIETY MECHANICAL ENGINEERS;
FORMERLY OF THE U. S. N. ENGINEER CORPS;
AUTHOR OF "MATERIALS OF ENGINEERING," "FRICTION AND LOST WORK," "HISTORY OF THE STEAM-ENGINE," "MANUAL OF STEAM-BOILERS," ETC. ETC.

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PREFACE.

This little treatise on methods of testing engines and boilers is an attempt to meet what has seemed to the Author a long-existing want. Hitherto, every engineer doing work of this kind has been compelled to do that work without a standard of reference, and the results of trials of engines and of boilers which have found their way into the record have been presented in such various ways as to be difficult of comparison, and such as to offer to the engineer desiring to do his work in an acceptable and permanently useful manner no generally accepted criterion. But the work of a committee of the American Society of Mechanical Engineers, of the German engineers, and of one or two individual and recognized authorities among experts, at later dates, has led to such a general concurrence among members of the profession that it is now possible to at least provisionally offer a system of testing both engines and steam-generators that may be accepted as satisfactory.

That this system will be steadily and constantly improved cannot be doubted, and the methods in vogue among the best practitioners to-day will not be precisely those in use among such experts a year hence; but the processes now adopted most generally will probably only be modified in detail, and the improvements will now mainly consist, it may be safely presumed, in the application of the most recent and accurate methods of precise measurement, as customarily applied in laboratories, to the determination of the quantities sought in such trials. The main outline of the scheme of trials to-day will be the substantial representative of similar operations later. A time has thus arrived when it is possible to put in permanent

form, and to publish for general use, the schemes of trials which have just taken definite shape. This is what the Author has attempted to do in this treatise.

It is proposed to present here those methods of trial of heat-engines which have become standard; to exhibit the processes of their application; to describe the best forms of apparatus used to date in conducting them and in securing the data sought; to illustrate their use and their various capabilities; and, finally, to present examples of the reports made by distinguished engineers on important work of this character, and thus to give good exemplars of their form, and of the data and results deduced from them in the case of the better classes of machinery and apparatus. It is intended that the apparatus in common use shall be described, as well as the method of its application to its purposes; its capabilities in the direct or indirect procurement of data; the processes of computation of the latter; the method of arrangement and tabulation of results; and the final compilation and report on the essential quantities which are required to give basis for the determination of the economy and efficiency, physical or commercial, of the machinery employed for the development of power.

The system of boiler-trial described is that proposed by the committee above referred to, and which has since become conventionally standard throughout the United States, and largely abroad. It is more complete and more satisfactory, in the opinion of the Author, than any other yet published, and seems to have been found sufficient to meet every ordinary requirement. The special precautions advised by the several experts on the committee are also quoted, and the forms of blanks and records found best adapted for use in such work are given. Some consideration is given to the methods of determining the character and value of the steam supplied by the boiler; and the forms of calorimeter generally used are described. It was shown by the results of a trial made by the Author for a committee of the American Institute, in 1871, that the best boilers, worked under ordinarily satisfactory conditions, give practically dry steam; but the necessity is none the less imperative to make certain, at every important trial, that this is the case with the boilers under test. A trial made without such determination of the quality

of the steam would, to-day, be regarded by the most expert among the profession as comparatively valueless.

The text adhered to in this account of the standard boiler-trial is that of the committee, as published in the transactions of the society. References are given in all cases in which the subject is such as would justify the reader in looking up original sources of information.

The chapter on the indicator is a brief and simple account of that wonderful instrument and its capabilities, as well as a description of the usual and best ways of handling it. No attempt has been made to elaborate to any great extent the study of the diagram; but the better forms of diagram, and those which are most likely to be met with in the best, as well as those taken from some defective, engines, are illustrated. For further information upon this exhaustless subject, the reader will consult the special treatises on the steam-engine indicator, of which a number exist, each, from that of Porter, the first workmanlike presentation of the subject, up to the latest extensive work published, having its own special field and its own peculiar characteristics. All give information of real value. It has been endeavored to give some idea of the shape and of the signification of the more usual and familiar forms of "card," and to show just how they bear upon the adjustment, the proportions, and the working of the engine; while singular and rare forms, which, however, so greatly interest every engineer, are generally left to be described by the writers of the special treatises on the indicator. In this respect, the published writings of several specialists of great experience and ability in the handling of the instrument will be found peculiarly rich.

In the chapter on the measurement and computations of the indicator diagram will be found a description of the methods usually considered best and most exact, and of the processes leading to the more important of the results attainable by the use of the instrument. These are mainly well known and standard among the best practitioners; but a few are of recent application and comparatively unknown, and are here for the first time introduced into a treatise of this kind. Such are the applications of the chronograph, applied, probably for the first time, in the manner exhibited in the frontispiece, by Professor W. A.

Anthony, and used for various purposes by a few engineers when seeking to obtain precise measurements of minute and rapid variations of engine-speed. The use of the timing-fork, employed for some years past in the work of the Author, is another novel and hitherto undescribed method of securing a record of the variations of velocity of parts of the engine, and one which promises to be of great value in investigations of the character described in illustration of its use in this chapter.

The application of the Prony or dynamometric brake is still another no less important and rarely described process of securing what are now universally recognized as essential measurements in the determination of the real efficiency of the engine. The several best-known modern forms of brake are here described, and their theory and the mode of their application are given in sufficient detail, it is hoped, to enable any one to properly apply them. The transmitting dynamometer is also often now used and finds its place in the text.

Standard methods of engine-trial are as essential to the satisfactory work of the engineer as are standard boiler-trials. There exists no such precisely formulated standard for engine-as for boiler-trials; but the text includes descriptions of such methods as are, in the opinion of the best authorities, likely to give, on the whole, most satisfactory, complete, and exact results. The schemes of competitive trials as customarily conducted are presented, and examples of special methods and results are given.

The work is completed by the introduction of a series of valuable reports written by a number of the ablest members of the profession as exemplars and models of most admirable summaries of data, and of conclusions derived from their study and from the computations made therefrom. One example of each of the most important classes of engine is studied in this manner; and the series should enable any engineer unfamiliar with such work by earlier experience to secure results thoroughly satisfactory to himself and to his clients. All essential constants and tables are given in the Appendix, and should others be desired in special cases, they can be readily found—usually in the table-books which cover the desk of every engineer and every student of engineering.

The Author has sought to compile a concise, accurate, and satisfactorily complete account of the apparatus and methods of the time, as familiar to the most experienced and accurate specialists. But even as the book is going through the press, new methods are being devised and old ones improved; while new instruments are being invented and the familiar apparatus of physical measurements are being applied to new purposes. Only the careful watch of current periodical literature will enable the engineer, young or old, to keep fully up with the progress of the age; it is nevertheless hoped that this compilation may prove of service to many, both young and old, and may be found to include enough of the most modern and the best practice to enable its reader to attain, in his own work, satisfactory results both as to accuracy and completeness.

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ENGINE AND BOILER TRIALS.

CHAPTER I.

REASONS AND PURPOSES.

1. **The Purpose of Test-trials** of engines and boilers, is, commonly, the verification of the claims of the builder to complete fulfilment of his contract, and more especially as to the power and economical working of his apparatus. Whenever a motor, whether an air, a gas, or a steam or other vapor engine, is constructed for a proposing purchaser and user, the builder is expected to bind himself by a carefully drawn contract to supply apparatus capable of developing a stated amount of power and with a specified consumption of fuel, and, sometimes, of other supplies. A test-trial is demanded, when the machine is set up and in normal operation, to ascertain whether such contract and its specifications have been completely fulfilled.

In other cases, a trial is made to satisfy the proprietor that his machinery is doing good work; in still other instances, he desires to ascertain whether variations of the usual methods of operation and rules of management may be expected to give improved results; sometimes, he desires to test the skill of his men, or the character of the fuel employed. In all cases, whatever the main purpose of the operation, certain data are sought to be obtained as a basis for computation of the results needed, either to give a measure of the power and efficiency of the

machinery, or a means of comparison with other apparatus of similar character and known excellence.

A complete trial of engine and boiler involves the determination of the quantity of energy stored in potential form in the fuel; the amount liberated by combustion in available form; the proportion and the quantity taken up by the boiler; the amount stored in the steam, and in any water taken up by it, and transferred to the engine; and the distribution at the engine into useful and lost work, and wasted heat. The methods of computation of these quantities will be given presently.

The purposes and methods of such trials are thus the exact and unquestionable determination of one or several of the efficiencies of the engine—or the boiler—and these methods are usually intended to be such as will give scientifically accurate measures of the heat, the steam, the feed-water, and the energy supplied to the system; the heat, steam, and energy reaching the engine; the power developed; the distribution, usefully and wastefully, of heat, energy, or power, or of all; the power and the thermodynamic and the actual efficiency of the engine considered as a heat-engine; and also the efficiency of the engine considered as a train of mechanism, *i.e.* as a machine. It is not always essential that all these determinations shall be made, or that such as are made shall be rigidly exact. Trials are often made which give partial results, and by methods which are only approximate and sometimes but roughly approximate, if judged from the standpoint of experimental science. As in all engineering work, the ultimate gauge of expediency, as judged from a financial standpoint and with an eye to a final summation of results, determines the extent to which the engineer is justified in giving his time and incurring expense in making steam-engine and boiler trials. On extensive contracts and important and costly work, all the resources of physical science and engineering practice are applied; in minor matters, but little expense or labor is deemed justifiable.

2. Specifications of Performance, and, often, a guarantee, with forfeiture in case of non-fulfilment, should form a part of

the contract; and those assurances of efficiency should be so exact and definite that no question can arise as to their meaning and fair interpretation, when the time arrives for their verification. The customary forms of such specifications are now fairly well settled, and the usual methods of comparison and verification will be exhibited and exemplified. When no such specification exists, it is assumed that the maker is bound to do reasonably good work and to assure to the buyers reasonably good economical performance. Obvious and unquestionable delinquency, as shown by test-trial, relieves the buyer of every responsibility not specifically and unqualifiedly assumed, and throws it upon the constructor and vendor.

Engine Duty is commonly the technical measure of the efficiency of the engine as determined by the cost of the work done in fuel consumed. The "horse-power," taken in British measure as 33,000 foot-pounds per minute or 1,980,000 per hour, demands the transformation of the equivalent amount of heat into work each minute or hour; which quantity should be supplied by one-fourth of a pound of good fuel, or less than 2½ pounds of steam, as worked in the perfect, ideal engine having an efficiency, unity. The actual consumption of energy derived from the boiler, as will be seen later, is rarely less than eight or ten times these amounts.

Pumping-engines are commonly rated by the work done by the consumption of a specified weight of fuel, as one hundred pounds. A duty of 100,000,000 foot-pounds, on this basis, would correspond to a consumption of 1.98 pounds of fuel per horse-power per hour. Mr. Sherman, assuming 90 per cent. as the efficiency of machine, including pumps as well as engine, obtains the figures in the following table (page 4).*

Mr. Emery has compared steam-engines of various kinds on the assumption that the boiler is capable of absorbing 10,000 heat-units per pound of coal consumed. This corresponds to an evaporation of 8.99 pounds of water at 80 pounds; 9.03 pounds at 60 pounds; or, 9.08 pounds at 40 pounds gauge-

* Trans. Am. Water-works Assoc., 1885.

DUTY OF PUMPING-ENGINES.

ONE MILLION GALLONS OF WATER IN 24 HOURS RAISED 200 FEET. DUTY AND QUALITY OF FUEL PER HOUR PER HORSE-POWER.

Duty.	Lbs. coal per million gallons.	Lbs. coal per hour per H. P.	Duty.	Lbs. coal per million gallons.	Lbs. coal per hour per H. P.	Duty.	Lbs. coal per million gallons.	Lbs. coal per hour per H. P.	Duty.	Lbs. coal per million gallons.	Lbs. coal per hour per H. P.
30	5560	5.94	51	3271	3.49	71	2349	2.51	[illegible]	1833	1.96
31	5380	5.75	52	3208	3.43	72	2317	2.48	92	1813	1.94
32	5212	5.57	53	3147	3.39	73	2285	2.44	93	1793	1.92
33	5054	5.40	54	3089	3.30	74	2254	2.41	94	1774	1.90
34	4906	5.24	55	3033	3.24	75	2224	2.38	95	1756	1.88
35	4766	5.09	56	2979	3.18	76	2195	2.35	96	1738	1.86
36	4634	4.95	57	2926	3.13	77	2166	2.31	97	1720	1.84
37	4508	4.82	58	2876	3.07	78	2138	2.28	98	1702	1.82
38	4390	4.69	59	2827	3.02	79	2111	2.26	99	1685	1.80
39	4276	4.57	60	2780	2.97	80	2085	2.23	100	1668	1.78
40	4170	4.45	61	2734	2.92	81	2059	2.20	101	1651	1.76
41	4068	4.35	62	2690	2.87	82	2034	2.17	102	1635	1.75
42	3972	4.24	63	2648	2.83	83	2009	2.15	103	1619	1.73
43	3880	4.14	64	2606	2.78	84	1986	2.12	104	1604	1.71
44	3790	4.05	65	2566	2.74	85	1962	2.10	105	1589	1.70
45	3707	3.95	66	2527	2.70	86	1940	2.07	106	1573	1.68
46	3626	3.87	67	2490	2.66	87	1917	2.05	107	1559	1.67
47	3549	3.79	68	2453	2.62	88	1895	2.02	108	1544	1.65
48	3475	3.71	69	2417	2.58	89	1874	2.00	109	1530	1.63
49	3404	3.63	70	2383	2.55	90	1853	1.98	110	1516	1.62
50	3336	3.56									

pressure from a temperature of 100° F. in each case. Ten thousand heat-units per pound of coal is equivalent to one million heat-units per 100 pounds of coal, and as the duty of pumping-engines is conventionally expressed in millions of foot-pounds per 100 pounds of coal, it follows on the basis presented that *the number of foot-pounds per heat-unit represents also the number of millions of foot-pounds duty per* 100 *pounds of coal.** The performance of all kinds of steam-engines may be readily compared on this basis. Ten thousand heat-units per pound of coal represent an efficiency of only (10,000 × 100 ÷ 14,500 =) 69 per cent. of the calorific value of pure carbon and of the best anthracite; so that ordinarily more than (100 — 69 =) 31 per cent. of the heat in the fuel is carried up the chimney. The mechanical equivalent of one heat-unit is *772* foot-pounds, which, on the basis above, correspond to a duty of *772* millions of foot-pounds per 100 pounds of coal. The most economical

* Centennial Report, Group XX, 1876.

steam-engines have been claimed to give as a maximum only about 130 millions, on the same basis, equivalent to an ultimate efficiency of ($130 \times 100 \div 772 =$) 16.84 per cent. of the heat in the steam, and but ($16.84 \times .69 =$) 11.62 per cent. of the calorific value of the fuel. If

$D =$ duty in foot-pounds per 100 pounds of coal,
$H =$ the height of lift per gauge,
$T =$ the initial and t the final temperatures, respectively,

then

$$D = \frac{1\,000\,000}{T - t + .0013H}.$$

The standard taken by British engineers in measuring duty is usually the number of pounds raised one foot either by 94 pounds of coal, or by one "hundred-weight" (112 pounds). On this basis, the duty is computed from the ascertained weight of fuel per I. H. P. per hour, thus:

$D =$ duty in millions;
$F =$ weight of fuel per I. H. P. per hour;

$D = \dfrac{186.12}{F}$, where the unit is 94 lbs.;

$D = \dfrac{221.76}{F}$, where in cwts.

3. **The Various Objects, in Detail,** sought in test-trials are determined in part by the specific purpose of the trial; but, in nearly every case, they require the measurement of power obtained and of cost of obtaining it, expressed either in money or in fuel consumed; and this means the exact weighing of fuel, the measurement of water used, the determination of the quality of the fuel and of the steam made, and its quantity, and the distribution of the stored energy, by the engine, in useful and in wasteful directions. Every quantity must be exactly measured which has importance in relation to the question at issue, and the data collected must be secured in such manner that their

magnitudes may be readily introduced into the computations and may, if question arises, be readily checked and verified. The methods of determination and of record thus become matters of importance and careful study, and the application of the greatest possible ingenuity and skill are demanded in the effort to devise an acceptable and reliable system. In studying the efficiency of capital, it is first necessary to consider the elements of cost of power.

The Annual Cost of Steam-power consists:

(1) Of certain expenses, which, in any given case, are usually invariable, whether the work is done by a large engine with high ratio of expansion and small boilers, or with a smaller engine working at a low rate of expansion and with larger boilers. These are usually: rent of building or interest on cost, taxes, repairs, etc., etc., of structure and location, the engineer's salary, and sometimes all or part of the fireman's or stoker's, also sundry minor expenses or a part of each of other expenses which, as a whole, are variable. Both of the latter classes may usually be neglected in solving the problem here first considered.

(2) The interest on first cost of engine in place, the cost of repairs, and a sum which measures the depreciation in value of the machine due to its natural wear, or to its decreasing value in presence of changes that finally compel the substitution for it of an improved engine. Oil, waste, and other engineer's stores fall under this head. These items are variable with size and style of engine.

(3) The expenses of supplying the engine with steam. These are:

(*a*) The cost on fuel account of the steam supplied, and which includes, also, the cost of steam condensed *en route* to the engines and wasted by cylinder condensation and leakage, as well as that actually utilized. This total quantity of steam greatly exceeds that actually used in the production of power by simple transformation of heat-energy. This item varies with the efficiency of engine and size of boiler demanded.

(*b*) The account of interest on cost of boilers in place and of their appurtenances, rent of boiler-room, depreciation, repairs,

and insurance, which latter account is wholly chargeable to boilers. This is also variable with size of boilers.

(*c*) Cost of attendance in excess of the costs included in the constant quantity in item (1) and variable with size of boiler or quantity of steam demanded.

The salary of the engineer is usually not chargeable to either engine or boiler; his position is one of supervision over the whole apparatus, and a good engineer generally keeps the closest watch over the boilers. The engine can usually be trusted, much of the time, to take care of itself. With small engines, the engineer is also the fireman. With large engines, the number of regular firemen—or, at least, the number in excess of one attendant—may be taken as proportional to the quantity of steam demanded when working at ordinary power, and with very large marine engines the same remark may sometimes apply to engine-room attendance.

The object of engine and boiler trials is to determine the magnitude of those quantities in item (3) *a* which can only be determined by direct and careful measurement.

The plan of this manual comprehends a study of the accepted methods of trial of engines and boilers, of the apparatus employed, the best methods of use, and the processes of determination of exact results and reliable conclusions. The various purposes of these investigations will be described and defined, and the quantities of which measures are to be sought will be specified; the various apparatus used in the work will be enumerated, and the general plan of the trials to be studied laid down. The instruments generally employed will be described fully and their proper manipulation shown. Methods of correction, of standardization, and of elimination of errors of observation and record will be indicated, and the character of all essential operations of computation exhibited. The usually practised systems of test-trial will be described, and the work illustrated by the presentation of sample reports and of results.

4. **The Maker and the Methods of the Trial** are usually subjects of stipulation in the contract. In some cases the parties to the contract merely agree that any question arising shall

be left to the arbitrament of a board of three competent men; one appointed by each, and the third by the first two appointees. In other cases, it is agreed that a known and reputable expert shall either conduct the trial or shall direct the appointments. Most frequently, when the work is very important, the contract provides for a board of three experts, who are usually named in the instrument, and also prescribes the method of conducting the trial. Those who are chosen for so important a duty should always be engineers of known ability, integrity, energy, and experience; whose professional practice has afforded them opportunities to become familiar with the best methods and has given them skill in their employment. Standard and well known and universally accepted methods should be prescribed whenever possible.

Earlier methods and authorities should be studied by those who desire to trace the progress of the art of engine-testing; but it will be found that very little has been done to reduce the several processes to exact form, and to a system, until very recently. A commission of eminent German engineers began such a codification of current practice, and a committee of the American Society of Mechanical Engineers has established a standard for tests of boilers. Various practitioners in this country and abroad, and the managing boards of public exhibitions and competitions, have gradually come to substantial agreement as to systems, and as to the methods which it is the purpose of this work to describe and which may be now taken as representing the most generally approved and recent practice.

5. **The Character of Report** demanded of the engineers conducting the trial is determined by the purpose of their work. In every case, it should be simple, easily comprehended, so far as that may be possible, by a non-expert reader; and it should give all essential data, processes, and results, in such manner that, in case they are called in question, they may be verified completely and with certainty. The matter sought to be settled should be defined with precision and clearness, and the whole statement should be as concise and as absolutely free from irrelevant matter as possible. In the endeavor to explain

presumably obscure points, even, opportunity will be found for the exercise of a good judgment and great discretion. Good examples of the best forms of report will be presented later.

Recent standard methods have approximated very closely to the exactness and accuracy of the processes of the physicist or the chemist. In fact, the apparatus and processes of these investigators are now adopted by the practitioner, and the young engineer himself is now almost invariably trained by the physicist and chemist, in all the exact methods of the laboratory, in his preliminary professional studies in the technical school. The determination of temperatures and the ascertainment of weights are conducted by all the accepted methods of exact scientific measurement, and the analysis of fuels, of gases, and of ashes is bringing into play and application some of the most interesting and accurate operations of the chemical laboratory.

6. The Apparatus employed in test-trials of engines and boilers should be made and standardized with all the care exacted in any other department of physical research. Instruments of every class used in such investigations should be carefully selected, and from the stock of the best makers; they should be inspected, tested, and compared with standard instruments of known excellence and accuracy, and any permanent errors in their operation and method of application should be recorded with every possible care. Scales should be compared with the legal standards; tanks and other volume-measures should themselves be very nicely measured; thermometers should be calibrated; indicators should be tested, hot as well as cold; and dynamometers should be measured up with similar accuracy.

7. Methods of Application of the instruments used are commonly well determined by experience and settled by custom. Such serious variations of result may be due to error in this matter that the study of the effects of differences of practice becomes an important part of the task of the engineer-expert. A fault in location of a thermometer, or in the setting up of an indicator, may produce quite sensible, and even serious, differences in the magnitude of data so obtained.

8. The Data Needed and Computations required in the determination of the character and performance of any engine or boiler, or of both, are obtained by means of continuous or closely successive readings from all instruments employed, so taken and recorded as to give a substantially exact record of the whole period over which it has been concluded to extend the trial. These observations must be sufficiently frequent and numerous to permit the computation of very accurate averages, and, where graphical methods are, as is now common, adopted, to permit the construction of smooth and complete curves showing the whole period of the trial. Every continuous process should thus be capable of representation by a similarly continuous record, and in such manner that every computation required may be readily effected. The results of computation will then furnish accurate numerical measures of every quantity needed to determine whether the contract has been fully complied with

9. Trials to Determine Economy and Efficiency are most common, and are generally of most importance. Few contracts for important steam apparatus are made which do not include a specification of the degree of economy demanded in the use of both steam and of fuel, and often of a system of test-trial as well. In such cases, it becomes necessary for the engineers conducting the trial to ascertain with precision the quantity of fuel, of steam, or of heat-energy supplied to the engine, the amount of energy converted into the mechanical form, the proportion of heat and of mechanical power wasted, the methods and extent of waste, in full detail, and the power applied by the machine to such useful purpose as it is designed to subserve. The measure of the benefit received by the user of the engine is the useful work performed; the measure of its cost to him is what he pays in fuel, steam, or heat-energy, and the money equivalent of this supply, and of all incidental costs, such as rent, attendance, wear and tear, insurance and taxes, and depreciation. A comparison of the mean continuous cost with the average value of power supplied for useful work exhibits the real value of the machinery to its purchaser and user.

Such a trial is only complete when it determines accurately the following:

EXPENDITURES: Quantities and Costs.	RECEIPTS: Quantities and Costs.
Fuel;	Useful Work;
or	Wasted Work and Heat:
Steam;	(*a*) Friction of Engine;
or	(*b*) Heat lost externally;
Heat-energy;	(*c*) Heat lost internally and
supplied by the user.	rejected from the system.

10. Steam-boiler Efficiency is not difficult of definition when the nature of the quantity to be measured is itself first understood. There are, however, as will be presently seen, several different efficiencies of the steam-boiler, as of the steam-engine; and it is important that each be distinctly defined before a study of either, or of total efficiency, can be made. In general, it may be said that efficiency is measured by the ratio, in common or similar and definitely related terms, of a result produced to the cost of its production. As, in the study of the steam-engine, either efficiency is measured by the ratio of work done in the specified manner to the work or work-equivalent expended in doing it; so, in the case of the steam-boiler, either efficiency is measured by the ratio of a heat-effect, or its equivalent, to the quantity of heat, actual or latent, paid for its accomplishment.

In some cases it is not practicable to thus establish a numerical value of an efficiency; and it can only be shown that efficiency, in the sense of quantity of result compared with magnitude of means used, is increased or decreased by the operation of defined phenomena, or by conditions which can be specified. A common measure cannot always be found, or an exact law of relation established.

Increasing steam-pressure gives increasing economy up to a limit somewhere above customary pressures. The higher the pressure the greater the economic value of the steam in a steam-engine; but on the other hand, the lower the efficiency

of the boiler; and it is perfectly possible to reach a point at which the gain on the first score is more than counterbalanced by the loss on the second. Where the object sought is simply heating-power, the advantage lies, on the whole, on the side of low pressures.

The measure of efficiency of boilers is commonly a ratio of heat applied to a defined purpose or obtained in store, in a stated form, to the total quantity of heat from which it has been saved, another part having been diverted to other purposes, and, for the use considered, wasted. Thus, a given quantity of heat being stored as potential energy of chemical action in fuel, some proportion of that energy is received at the steam-engine when that fuel is burned under a steam-boiler; the ratio of these two quantities—always a fraction and often small—is the *total* efficiency of the whole apparatus employed in the combustion of fuel, the transfer of heat-energy to the fluid in which it is stored, and its further transfer to the point at which it is usefully applied by transformation into mechanical energy and work.

The efficiency of combustion thus measures the ratio of the available heat-energy of the fuel to that set free by its union with oxygen, and is less than unity in the proportion in which the combustible portion of the fuel escapes such chemical change or is imperfectly burned, as when a part of the fuel falls into the ash pit, is imbedded in clinker, or remains on the grate when the fire is extinguished; or as when carbon is only oxidized to carbon monoxide instead of being completely burned into dioxide. In well-managed furnaces the value of this efficiency approaches unity; it ought not to fall below 0.90, probably, in any ordinary case.

The efficiency of transfer of heat similarly measures the ratio of heat received from the furnace by the boiler to that produced by combustion. That not transferred to the boiler is either sent up the chimney, where it is, in a certain degree, useful in producing draught, or it is lost by conduction and radiation to surrounding bodies. In good examples, the value of this ratio exceeds 0.75, and it should not usually fall under

fifty or sixty per cent. Its best value depends on considerations, however, to be hereafter stated, and it is not always desirable that it should have the highest value possible, or approximate unity.

The net efficiency of boiler is the continued product of all efficiencies.

11 **Efficiency of Heating-Surface** measures the ratio of actual amount of heat transmitted across such surface to the total quantity available for such application; in steam-boilers it is the ratio of the quantity of heat utilized in heating and vaporizing the fluid to the total which is produced by the furnace, the unutilized heat being wasted by conduction and radiation to other bodies, or sent up the chimney. An expression was found by Rankine which has been found to give very satisfactory results when properly used in application to the ordinary work of steam-boilers. This expression may be derived as below.

Let w be the weight of furnace-gases discharged per hour, $T - t$ the difference between the temperatures of gas and water on opposite sides of any part of the plate on the elementary area dS, C the specific heat of the gas, and let q be the quantity of heat passing across unity of area in unity of time for a difference in temperature $T - t$, in other words, the "rate of conduction" per unit of area per hour.

The quantity or heat transferred across the area dS is then eqal to qdS, and the fall of temperature of gas must be this quantity divided by the product of the weight, w, and specific heat, C, of the gas from which the heat is derived,

$$\frac{qdS}{Cw} = -dT;$$

and the gas flows on to the next elementary area and beyond, surrendering its heat as it goes, until it finally leaves the absorbing surface and enters the chimney-flue.

If T_1 and T_2 are the initial and final temperatures of the gas, and t the temperature of the water entering the boiler, the

heat produced, Q_1, and that wasted, Q_2, per hour, are respectively measured by

$$Q_1 = Cw(T_1 - t) : Q_2 = Cw(T_2 - t), \text{ nearly;}$$

while the efficiency of the heating-surface is measured by the ratio of total heat to absorbed heat; or, if the feed enters at atmospheric temperature, or nearly so, by

$$\frac{Q_1 - Q_2}{Q_1} = \frac{T_1 - T_2}{T_1 - t_1}, \text{ nearly.}$$

For the every-day work of the engineer dealing with familiar forms of steam-boiler under the ordinary conditions of daily operation, it is often most satisfactory to make use of empirical expressions derived from experiments conducted with boilers of similar character and under similar conditions of use. The expression elsewhere given as constructed by Rankine for efficiency of furnace is a rationally produced algebraic statement of a law, for example, which may also be approximated with quite sufficient accuracy, by direct determination, for any specified case. Should an error be made in assigning the correct coefficient, in the former, it may produce errors of serious magnitude; if the empirical expression be applied to other than the conditions on which it was based, errors will certainly arise which may become important.

But that expression (§ 11) and other formulas which are more or less empirical may be relied upon wherever the conditions under which it is proposed to apply them are substantially the same as those under which they were constructed and those which furnished the constants which enter into them. Where these conditions are precisely the same, empirical expressions like those of Emery (p. 19) and others are often considered even more satisfactory than any rational forms yet obtained; the reason of this fact being that the actual conditions of operation are usually not precisely those assumed by the author of the formula.

The heat utilized, $Cw(T_1 - T_2)$, is also equal to that absorbed and transmitted, qdS:

$$\int qdS = Cw(T_1 - T_2) \text{ and } \frac{S}{Cw} = \int_{T_2}^{T_1} \frac{dT}{q}.$$

The value of q has been found to be well represented by equation (4) of article 95, in which $q = \frac{Q}{At}$, and hence $q = \frac{(T_1 - T_2)^2}{a}$; and thus

$$\frac{S}{Cw} = \int_{T_2}^{T_1} \frac{dT}{q} = a\int_{T_2}^{T_1} \frac{dT}{(T-t)^2}.$$

Assume $(T - t) = x$, then

$$\frac{S}{aCw} = \int_{T_2}^{T_1} \frac{d(T-t)}{(T-t)^2} = \int_{T_2}^{T_1} x^{-2}dx$$

$$= \Big[-x^{-1}\Big]_{T_2}^{T_1} = \Big[\frac{1}{T-t}\Big]_{T_2}^{T_1};$$

$$\therefore \frac{S}{aCw} = \frac{1}{T_2 - t} - \frac{1}{T_1 - t} = \frac{(T_1 - t) - (T_2 - t)}{(T_1 - t)(T_2 - t)};$$

and the efficiency becomes

$$E = \frac{T_1 - T_2}{T_1 - t} = \frac{S}{aCw}(T_2 - t).$$

Then, since

$$\frac{T_1 - t}{T_2 - t} = \frac{S(T_1 - t) + 1}{aCw} = \frac{S(T_1 - t) + aCw}{aCw},$$

and

$$\frac{(T_1 - t) - (T_2 - t)}{T_2 - t} = \frac{T_1 - T_2}{T_1 - t},$$

$$E = \frac{T_1 - T_2}{T_1 - t} = \frac{S(T_1 - t)}{S(T_1 - t) + aCw}.$$

If the total heat absorbed per hour be taken as H,

$$H = Cw(T_1 - t); \quad T_1 - t = \frac{H}{Cw};$$

and a simplified expression,

$$E = \frac{S}{S + \frac{aC^2w^2}{H}}$$

is obtained, in which Cw may be taken as proportional to the weight of air supplied or of fuel burned, and H as proportional to the same quantity. Thus if F is the weight of fuel burned in the given time, on unity of grate-area, the efficiency may be expressed as

$$E = \frac{BS}{S + AF} = \frac{B}{1 + AR},$$

which is the formula sought. A and B are constants to be obtained by experiment for the special type of boiler to be considered.

When S and F represent respectively the number of square feet of heating-surface per square foot of grate in any boiler, and the number of pounds of fuel burned as the square foot of grate per hour, and $R = \frac{F}{S}$, the values of A and B, as given by Rankine,* are as follows:

* Steam-engine, p. 294.

Boiler Type.		*A.*	*B.*
Class 1.	Best convection, chimney draught......	0.5	1.00
" 2.	Ordinary " " "	0.5	0.90
" 3.	Best " forced "	0.3	1.00
" 4.	Ordinary " " "	0.3	0.95

These constants are derived from experience with good fast-burning bituminous coals; for anthracites of good quality the Author has usually found the following values more in accordance with good practice:

Boiler Type.	*A.*	*B.*
Class 1....................................	0.5	0.90
" 2....................................	0.5	0.80
" 3....................................	0.3	0 90
" 4....................................	0.3	0.85

When feed-water heaters are used, or superheaters are employed, their surface should be included in the area S. The formula assumes no loss by excess of air-supply. Where such excess is noted or anticipated, it may be allowed for by increasing the value of A in proportion to the square of the total quantity of air supplied. The following table presents values of efficiency for a wide range of practice:

EFFICIENCY OF BOILERS.

R.	Bituminous Coal. Class of Boiler.				Anthracite Coal. Class of Boiler.			
	I.	II.	III.	IV.	I.	II.	III.	IV.
10	0.16	0.15	0.25	0.22	0.14	0.14	0.23	0.20
4	0.33	0.31	0.45	0.43	0.30	0.28	0.40	0.39
2	0.50	0.46	0.62	0.59	0.45	0.50	0.56	0.53
1	0.66	0.61	0.77	0.73	0.60	0.55	0.70	0.66
0.80	0.71	0.65	0.81	0.77	0.64	0.59	0.73	0.69
0.67	0.75	0.69	0.83	0.79	0.67	0.63	0.75	0.72
0.50	0.80	0.73	0.87	0.83	0.72	0.65	0.78	0.75
0.40	0.83	0.76	0.89	0.85	0.75	0.68	0.80	0.77
0.333	0.86	0.80	0.90	0.86	0.77	0.72	0.81	0.78
0.167	0.93	0.85	0.95	0.90	0.84	0.77	0.86	0.81
0.111	0.95	0.87	0.97	0.92	0.86	0.78	0.88	0.83

These values have been found to agree well with practice up to rates of combustion exceeding 50 or 60 pounds per

square foot of grate-surface per hour, beyond which point the efficiency falls off. But agreement can only be expected where the combustion and air-supply are in accordance with the assumptions on which the formula is based.

The problem of the designer of steam-boilers often takes the form: Required to determine the area of heating-surface needed to secure a stated efficiency. In this case the formula above given must be transformed thus:

$$E = \frac{B}{1 + AR} = \frac{B}{1 + \frac{F}{S}};$$

$$S = \frac{F}{\frac{B}{E} - 1}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

$$R = \frac{S}{F} = \frac{1}{\frac{B}{E} - 1}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

from which expressions, the efficiency aimed at being given, the ratio of heating to grate-surface and the extent of heating-surface may be computed. As will be seen later, the question to what extent efficiency may be economically carried by extending heating-surface is one of the problems arising in designing boilers.

Mr. Emery finds that the results of extensive series of experiments on good forms of boiler may be represented by the empirical expression, for vertical tubular boilers,

$$E = \frac{46.045}{c + 3.016} - 1.067;$$

in which E is the ratio of weight of water evaporated into dry steam, "from and at the boiling-point," under atmospheric pressure, to weight of fuel used; and c is the weight of combustible consumed per square foot of heating surface per hour, in pounds.

For horizontal boilers,

$$E = \frac{27.287}{c + 2.04} + 0.824.$$

The evaporation is here the maximum practicable with good anthracite coal.

The maximum efficiency is given as

$$E_e = \frac{e}{E};$$

in which e is the observed evaporation, reduced to the standard basis. These relations are shown in the table on p. 20. For badly designed or mismanaged boilers deduction must be made from the efficiencies and evaporations here given, to the extent of ten per cent or more, according to magnitude of the defect.

In land boilers, it is customary to keep the rate of combustion per square foot of grate down to about eight pounds per hour, although it frequently rises to 10 and 12. In marine boilers this rate is increased to 12 and 16 pounds per square foot of grate per hour when anthracite coal is burned with natural draught, and to 20 pounds and upward per hour for bituminous coal. In a locomotive, however, with forced draught, 75 and 100 pounds of coal are burned per square foot of grate. Apparently no losses result from such variations in the size of the grate, and, in fact, it appears indisputably that with a reduced grate and forced draught the air-dilution is reduced and the evaporization therefore somewhat increased. It is reasonable to conclude, therefore, that when care is taken to insure perfect combustion by ordinary tests, the relative area of the grate upon which the coal is consumed does not affect the result, and economy depends under proper conditions upon the rate of combustion per unit of heating surface as stated.

12. **Effective Development, Transfer, and Storage of Heat,** in the best possible combination, is evidently what is demanded in the operation of the steam-boiler.

In securing complete combustion, an ample supply of air

PERFORMANCE OF BOILERS.*—EMERY.

1	2	3	4	5	6
c	E	cE	$E \div 15$	$34.52 + 1.2 \div E$	$34.52 \div cE$
Combustible consumed per square foot of heating surface.	Water evaporated at atmospheric pressure from temperature of 212°.		Ultimate efficiency.	Coal (with 1-6 refuse) per horse-power per hour.	Heating surface per horse-power.
	Per pound of combustible.	Per square foot of heating surface.		On basis that one horse-power requires 30 pounds of water per hour, evaporated at 70 pounds pressure from temperature of 100°, or 34.52 pounds at atmospheric pressure from temperature of 212°.	
Pounds.	Pounds.	Pounds.		Pounds.	Square Feet.
Minimum.	14.20	....	0.95	..	....
0.1	13.71	1.37	0.91	3.02	25.18
0.2	13.25	2.65	0.88	3.13	13.03
0.3	12.82	3.85	0.85	3.23	8.98
0.4	12.41	4.96	0.83	3.34	6.95
0.5	12.03	6.02	0.80	3.44	5.74
0.6	11.68	7.01	0.78	3.55	4.92
0.7	11.32	7.92	0.75	3.66	4.36
0.8	11.00	8.80	0.73	3.77	3.92
0.9	10.69	9.62	0.71	3.87	3.59
1.0	10.39	10.39	0.69	3.99	3.32
1.5	9.13	13.70	0.61	4.54	2.52
2.0	8.11	16.22	0.54	5.11	2.13
2.5	7.28	18.20	0.49	5.69	1.90
3.0	6.57	19.71	0.44	6.30	1.75
3.5	6.00	21.00	0.40	6.90	1.64
4.0	5.50	22.00	0.37	7.53	1.57
4.5	5.06	22.77	0.34	8.19	1.52
5.0	4.68	23.40	0.21	8.85	1.48

and its thorough intermixture with the combustible elements of the fuel are essential; for the second, high temperature of furnace, it is necessary that the air-supply shall not be in excess of that absolutely needed to give complete combustion. The efficiency of a furnace burning fuel completely is measured by

$$E = \frac{T - T'}{T - t},$$

* *Sci. Am. Supplement.* No. 687, p. 10,972.

in which E represents the ratio of heat utilized to the whole calorific value of the fuel; T is the furnace-temperature; T', the temperature of the chimney; and t, that of the external air. Hence the higher the furnace temperature and the lower that of the chimney, the greater the proportion of available heat.

It is further evident that, however perfect the combustion, no heat can be utilized if either the temperature of chimney approximates to that of the furnace, or if the temperature of the furnace is reduced by dilution approximately to that of the chimney. Concentration of heat in the furnace is secured, in some cases, by special expedients; as by heating the entering air, or, as in the Siemens gas-furnace, heating both the combustible gases and the supporter of combustion. Detached fire-brick furnaces have an advantage over the "fireboxes" of steam-boilers in their higher temperature; surrounding the fire with non-conducting and highly-heated surfaces is an effective method of securing more perfect combustion and high furnace-temperature.

In arranging heating-surface, the effort should be to impede the draught as little as possible, and so to place them that the circulation of water within the boiler should be free and rapid at every part reached by the hot gases.

The directions of circulation of water on the one side and of gas on the other side the sheet should, whenever possible, be opposite. The cold water should enter where the cooled gases leave, and the steam should be taken off farthest from that point. The temperature of chimney-gases has thus been reduced by actual experiment to less than 300° Fahr., and an efficiency equal to 0.75 to 0.80 the theoretical is attainable.

The extent of heating-surface simply, in all of the best forms of boiler, determines the efficiency, and the disposition of that surface in such boilers seldom affects it to any great extent. The area of heating-surface may also be varied within wide limits without greatly modifying efficiency. A ratio of 25 to 1 in flue and 30 to 1 in tubular boilers represents the relative area of heating and grate surfaces in the practice of the best-known builders. This proportion may be often settled by exact calculation.

The material of the boiler, as will be shown later, should be tough and ductile iron, or, better, a soft steel containing very little carbon and thoroughly homogeneous.

The factor of safety is very often too low. The boiler should be built strong enough to bear a pressure at least six times the proposed working-pressure; as the boiler grows weak with age, it should be occasionally tested to a pressure far above the working-pressure, which latter should be reduced gradually to keep within the bounds of safety. The factor of safety is seldom more than four in new boilers; and even this is reduced practically by the operation of the inspection laws.

Effective development of heat is secured primarily by the selection of good fuel, by which is usually meant fuel which consists, to the greatest possible extent, of available combustible material; but for the purposes of the engineer who designs the boiler, or of the owner for whom it is to be constructed, the real criterion of quality is the quantity of heat which the combustible, as burned in the furnace, will yield for any given sum of money expended in obtaining that heat. The cost of a fuel to the consumer consists, not simply of money paid for it to the dealer who supplies it, but also of cost of transportation and of placing in the grate, of removal of ash, of incidental expenses inseparable from its use, such as injury to boilers and other property, increased risks, and other such expenses, many if not most of which are very difficult of determination with any satisfactory decree of accuracy. Other things being equal, that fuel which gives the greatest quantity of available heat for the total money expenditure is that which permits most effective development in the sense here taken. Effective heat-development from any selected fuel is secured, as already stated, by its complete combustion in such manner as to give the highest possible temperature.

Effective transfer of heat is secured by such a form of steam-generator, and such extent and disposition of "heating-surfaces," as will most completely utilize the heat developed in the furnace and flues by causing it to flow, with the least possible loss, into the water and steam contained within the boiler; and this is effected by proper arrangement of surfaces absorb

ing heat from the gases and yielding it to the liquid as already generally described.

Effective storage of heat can be secured by providing large volumes of water and of steam, within which the heat transferred from the furnace and flues can be stored, and by carefully protecting the whole heated system from waste by conduction or radiation to adjacent bodies. Where the demand is steady, and the supply from the fuel fairly steady also, the amount stored need not be great, as the use of the reservoir is simply that of a regulator between furnace and engine or other apparatus receiving it; but where either supply or demand is variable, considerable storage capacity may be needed.

13. **Efficient Utilization of Heat** is as essential to the satisfactory working of any system of generation and application of heat as is efficient production, transfer, and storage. The mode of attaining maximum efficiency depends upon the nature of the demand and the method of expenditure; and the consideration of this subject in detail would be here out of place. In general it may be said that where the heat and steam are required for the impulsion of an engine, the higher the safe pressure and the practically attainable temperature at which the supply is effected, the more efficient the utilization of the heat. These limits of temperature and pressure are the higher as the actual working conditions are made the more closely to approximate to the ideal conditions prescribed by pure science.

Where heating simply, without transformation into work, is intended, the principal and only very important requisite, usually, is to provide such thorough protection for the system of transfer and use, that no wastes of importance can take place by radiation or conduction. The character of the steam made, as to humidity, is in this case comparatively unimportant; but in the preceding case it will be found essential that it should be always dry, and it is often much the better for being superheated considerably above the boiling-point due to its pressure.

The actual standing of the best steam-engine of the present time, as an efficient heat-engine, is really very high. The sources of loss are principally quite apart from the principles of design and construction, and even from the operation of the

machine; and it may be readily shown that, to secure any really important advance toward theoretical efficiency, a radical change of our methods must be adopted, and probably that we must throw aside the heat-engine in all its forms, and substitute for it some other apparatus by which we may utilize some mode of motion and of natural energy other than heat.

The very best classes of modern steam-engines very seldom consume less than two pounds (0.9 kilog.) of coal per horse-power per hour, and it is a good engine that works regularly on three pounds (1.37 kilog.).

The first-class steam-engine, therefore, yields less than 10 per cent of the work stored up in good fuel, and the average engine probably utilizes less than 5 per cent. A part of this loss is unavoidable, being due to natural conditions beyond the control of human power, while another portion is, to a considerable extent, controllable by the engineer or by the engine-driver. Scientific research has shown that the proportion of heat stored up in any fluid, which may be utilized by perfect mechanism, must be represented by a fraction, the numerator of which is the range of temperature of the fluid while doing useful work, and the denominator of which is the temperature of the fluid when entering the machine, measured from the "absolute zero."

Thus, steam, at a temperature of 320° Fahr., being taken into a perfect steam-engine, and doing work there until it is thrown into the condenser at 100° Fahr., would yield $\frac{320 - 100}{320 + 461} = 0.28 +$, or rather more than one fourth of the work which it should have received from each pound of fuel. The proportion of work that a non-condensing but otherwise perfect engine, using steam of 75 pounds (5 atmos.) pressure, could utilize would be $\frac{320 - 212}{320 + 461} = 0.14 = \frac{1}{7}$; and, while the perfect condensing engine would consume two thirds of a pound (0.3 kilog.) of good coal per hour, the perfect non-condensing engine would use $1\frac{1}{8}$ pounds (0.6 kilog.) per hour for each horse-power developed, the steam being taken into the engine and exhausted at the temperatures assumed above.

Also, were it possible to work steam down to the absolute zero of temperature, the perfect engine would require but 0.19 pound (0.09 kilog.) of similar fuel.

It may therefore be stated, with a close approximation to exactness, that of all the heat derived from the fuel about seven tenths is lost through the existence of natural conditions over which man can probably never expect to obtain control, two tenths are lost through imperfections in our apparatus, and only one tenth is utilized in even good engines. Boiler and engine are intended to be included when writing of the steam-engine above. In this combination a waste of probably two tenths at least of the heat derived from the fuel takes place in the boiler and steam-pipes, on the average, in the best of practice, and we are therefore only able to anticipate a possible saving of $0.2 \times 0.75 = 0.15$, about one sixth of the fuel now expended in our best class of engines, by improvements in the machine itself. The best steam-engine, apart from its boiler, therefore, has 0.85, about five sixths, of the efficiency of a perfect engine, and the remaining sixth is lost through waste of heat by radiation and conduction externally, by condensation within the cylinder, and by friction and other useless work done within itself. It is to improvement in these points that inventors must turn their attention if they would improve upon the best modern practice by changes in construction.

To attain further economy, after having perfected the machine in these particulars, they must contrive to use a fluid which they may work through a wider range of temperature, as has been attempted in air-engines by raising the upper limit of temperature, and in binary vapor engines by reaching toward a lower limit, or by working a fluid from a higher temperature than is now done down to the lowest possible temperature. The upper limit is fixed by the heat-resisting power of our materials of construction, and the lower by the mean temperature of objects on the surface of earth, being much lower at some seasons than at others. In the boiler the endeavor must be made to take up all the heat of combustion, sending the gases into the chimney at as low a temperature as possible, and securing in the furnace perfect combustion without excess of air-supply.

Good furnace management, to secure maximum heat-supply from the unit weight of fuel, is evidently as essential to economy and efficiency of steam production as choice of proper fuels.

In the management of the furnace the effort should be made to secure the best conditions for economy, and as nearly as possible perfect uniformity of those conditions. The fuel should be spread over the grate very evenly, and the tendency to burn irregularly, and especially into holes or thin spots, should be met by skilful "firing," or "stoking," as it is also termed, at such intervals as may by experience be found best. The smaller the coal, where anthracite is used, the thinner should be the fire; the stronger the draught, the thicker the bed of fuel, of whatever kind. With too thin a fire, the danger arises of excess of air-supply; with too heavy a fire, carbon monoxide (carbonic oxide) may be produced. In the former case combustion will be complete, but the heat generated will be distributed throughout the diluting excess of air, and thus rendered less available, and the efficiency of the furnace will be correspondingly reduced; while in the latter case a loss arises from incomplete combustion, and waste takes place by the passage of combustible gas up the chimney. The second is the less common cause of loss of the two, but both are liable to arise in almost any boiler, and we may even have both losses exhibited in the same boiler and at the same time. Successful working demands a very perfect mixture of the combustible with the supporter of combustion, and should this not be secured, serious waste will take place.

The appearance of smoke at the chimney-top is not always indicative of serious loss, nor is its non-appearance always proof of complete combustion. With soft coals and other fuels containing the hydrocarbons some smoke usually accompanies the best practically attainable conditions; anthracites, charcoal, and coke never produce true smoke. Attempts to improve the efficiency of a heat-generating apparatus by "burning the smoke" usually fail by introducing such an excess of air as to cause a loss exceeding that before experienced from the forma-

tion of smoke. Thorough intermixture of a minimum air-supply with the gases distilled from the fuel is the only means of attaining high efficiency.

In firing, or stoking, especial care should be taken to see that the sides and corners of the grate are properly attended to. Regulation of the fire is best secured by the careful adjustment of the damper. The manipulation of the furnace doors for this purpose is likely to cause waste. Liquid fuels are especially liable to waste by excessive air-supply, and gaseous fuel exhibits a peculiar liability to the opposite method of loss; both should be, if possible, even more carefully handled than any solid fuels.

14. Trials to Ascertain Power or Maximum Capacity to do useful work are often, perhaps usually, made under the same contract as are those to determine efficiency of engine. Steam machinery is commonly guaranteed, both as to economy and capacity. Such trials are sometimes made at the same time with efficiency-trials; but the maximum power of engines and boilers is seldom, if ever, that at which best economy is obtainable. A single trial is made when the power guaranteed is that of normal working and that for which the guarantee of economy is made by the contract. A trial for capacity simply, is one in which the power only need be measured, and its cost, unless specifically demanded, is not determined. The methods employed, so far as they go, are the same as in the preceding and more complete kind of trial.

The actual power of steam and of boilers evidently depends upon the efficiency of the method of application, and on the apparatus employed. The quantity of heat-energy supplied to the engine and yielded by the generator has been seen to be easily calculable by simply multiplying the quantity of heat given to the steam, by the fuel, by the mechanical equivalent of heat. The amount available as energy may be the total quantity so supplied, as when the steam is condensed in heating buildings or otherwise, and is returned as feed-water to the boilers; or it may be any less amount, according as the method of utilization is more or less effective. The tables given in the

Appendix furnish the data for calculation in any case in which the efficiency of transfer and of transformation is known. Where no constant value can be assumed for the efficiency of the system employed, it is sometimes, nevertheless, found to be important to establish a standard conventionally. Thus, in the calculation of available stored energy, as given in the Appendix, it was assumed that the steam would be expanded to atmospheric pressure. Similarly, convention has established the unit horse-power of steam-boilers, in order to afford a standard of comparison in test-trials, and to give a means of rating boilers by the designer, the builder, or the purchaser and user.

The operation of boilers occurs under a wide range of actual conditions the steam-pressure, the temperature of feed-water, the rate of combustion and of evaporization, and, in fact, every other variable condition, differing in any two trials to such an extent that direct comparison of the totals obtained, as a matter of information regarding the relative value of the boilers, or of the fuel used, becomes out of the question. It has hence gradually come to be the custom to reduce all results to the common standard of weight of water evaporated by the unit-weight of fuel, the evaporation being considered to have taken place at mean atmospheric pressure, and at the temperature due that pressure, the feed-water being also assumed to have been supplied at the same temperature. This, in technical language, is said to be the "equivalent evaporation from and at the boiling-point" (212° F., 100° Cent.). This standard has now become generally incorporated into the science and the practice of steam-engineering. The "Unit of Evaporation" is one pound of water at the boiling-point, evaporated into steam of the same temperature. This is equivalent to the utilization of 966, nearly, British thermal units per pound of water so evaporated. The economy of the boiler may thus be expressed by the number of units of evaporation obtained per pound of combustible.

Newcomen used steam of barely more than atmospheric pressure, and raised 105,000 pounds of water one foot high, with a pound of coal consumed. Smeaton raised the steam-pressure to eight pounds, and increased the duty to 120,000. Watt

started with a duty of double that of Newcomen, and raised it 320,000 foot-pounds per pound of coal, with steam at ten pounds. To-day, Cornish engines of the same general plan as those of Watt, but worked with forty to sixty pounds pressure, expanding three to six times, bring up the duty to 600,000 foot-pounds; while more modern compound engines have boilers carrying 150 pounds (ten atmospheres) above the normal air pressure, and the duty has been since raised to above 1,200,000 foot-pounds per pound of fuel used.

15. The Quantities Measured and Results sought to be secured are thus, in detail, as follows for any complete trial:

When the trial includes, as is most frequently the case, a trial of the boiler, the combined efficiency of boiler and engine being the final determination, arrangements must be made in advance to ascertain exactly the weights of fuel, gross and net, coal and ash for example; the weight of water supplied as "feed;" the weights, temperatures, and pressures of dry steam, and weight of entrained water; the temperatures of furnace, flues, and chimney; of superheating steam, if it be so heated; the power of the engine, gross and net; the friction of engine; the wastes by cylinder-condensation and otherwise; the steam-pressure in boiler and steam-chest; and the continually varying pressures in the working cylinder throughout the whole cycle, revolution by revolution, of the engine. Each of these quantities is measured at specified intervals, and a comparison of mean values of power usefully applied, and of expenditures made to produce it, gives the measure of the economy attained.

At the same time that the gross power developed by the action of the steam in the cylinder, the indicated power, is measured, the diagrams taken furnish the means of ascertaining precisely how the pressures and volumes of the steam simultaneously vary within the engine, and thus give a clue to, and usually a fairly exact determination of, the setting and motion of the valves and the extent to which the distribution of steam is such as will best conduce to economical working. These diagrams also enable the engineer to compute with considerable accuracy the volumes and weights of the steam at any

point, and at every point, in the stroke. A comparison of the quantities so calculated with the actual measures obtained at the boiler, or before the steam enters the cylinder, gives the measure of the quantity condensed in the cylinder, as the piston moves forward, and of the later re-evaporation. The cylinder-wastes are thus also determinable with a fair degree of accuracy. These "cards" also exhibit the amount of back-pressure, and this measures the resistances in the exhaust passages and at the condenser, if there be one, and thus afford a means of criticism of the design and construction of the engine in this respect. Similarly, the difference between the steam-pressures in the cylinder and in the steam-chest and the exhaust-chest, is a measure of the losses in the steam passages.

The usual rates of evaporation and the effect of varying the proportions of tubes has been well determined by the experiments of Isherwood and others.

The proportions of flues and tubes vary somewhat in practice; but it will be found seldom advisable to make tubes more than 50 or 60 diameters in length. Where the heating-surface consists principally of tubes, the efficiency will be found to vary with their length nearly as follows:

Length of tube (diameters)	60	50	40	30	20
Water per unit-weight of fuel	12	11	10	9	8

When the ratio of heating to grate area was 25 to 1, Isherwood found the evaporation to vary thus:

Fuel per hour	8	10	12	16	20	24
Evaporation	10.5	10.1	9 5	8.2	7.3	6.8

which series is represented by

$$W = \frac{21}{\sqrt[3]{F}}, \text{ nearly.}$$

Clark obtained with locomotives an equal evaporation with

Fuel (coke)	15	25	38	56	76	98	125	153
Ratio of H. S to G. S.	30	40	50	60	70	80	90	100

the evaporation being constant at 9 of water to 1 of fuel, which may be expressed by

$$S = 8\sqrt{F}, \text{ nearly,}$$

S being the ratio of the two areas and F the weight of coke burned on the unit of area of grate.

In estimating area of heating-surfaces the whole surface exposed to the hot-furnace gusts is reckoned. The formula for efficiency already given illustrates the progressive variation of the evaporative power with change of proportions of boiler.

The relation of size of boiler to quantity of steam demanded is one that occasionally becomes worthy of consideration. Where the steam is required for driving steam-engines it is very important that it should be thoroughly dry, and it is an advantage to moderately superheat it. Maximum economy cannot be attained where wet steam is used. A boiler attached to a steam-engine, and especially where fuel is costly and efficiency important, should have ample heating-surface, some superheating-surface if practicable, ample extent of water-surface area to permit free separation of steam and water, and large steam-space.

Steam employed for heating purposes is not necessarily dry; it may carry a large amount of water with it into the system of heating-coils or radiators, and yet give good results, if the latter are of large section. Where the pipes are of restricted area of section, however, wet steam flowing less freely than when dry or superheated, there may result such a retardation of flow and of circulation as may cause considerable increase of cost. This has been found sufficiently great, in some cases, to justify drying, and perhaps superheating, the exhaust-steam from engines where used for heating purposes. As a general rule, the boiler must be made a trifle larger to supply perfectly dry steam and do good work.

In the use of steam for heating purposes, one square foot of boiler-surface will supply from 7 to 10 square feet of radiating surface. Small boilers should be larger proportionately than large boilers. Each horse-power of boiler will supply from 250

to 350 feet of 1-in. steam-pipe, or 80 to 120 square feet of radiating surface.

Under ordinary conditions one horse-power will heat about

Brick dwellings, in blocks, as in cities.......	15,000 to 20,000 cub. ft.
" stores " "	10,000 " 15,000 " "
" dwellings exposed all around	10,000 " 15,000 " "
" mills, shops, factories, etc.....	7,000 " 10,000 " "
Wooden dwellings, exposed.......	7,000 " 10,000 " "
Foundries and wooden shops...............	6,000 " 10,000 "
Exhibition buildings, largely glass, etc.	4,000 " 15,000 " "

The system of heating mills and manufactories by means of pipes placed overhead is recommended.

The air required for ventilation is usually warmed by the "indirect" system of radiation, the current passing through boxes or chambers in which a sufficient amount of pipe is coiled to heat it well. From 5 to 15 cubic feet per individual per minute are allowed, the former in crowded halls, the latter in dwellings, and about one-tenth as much for each gas-burner or lamp.

Small engines, according to Buel, demand steam, ordinarily, as below:

Pressure of Steam in Boiler, by Gauge.	Pounds of Water per effective Horse-power per Hour.	Pressure of Steam in Boiler by Gauge.	Pounds of Water per effective Horse-power per Hour.
10	118	60	75
15	111	70	71
20	105	80	68
25	100	90	65
30	93	100	63
40	84	120	61
50	79	150	58

Pressures lower than 60 pounds are not usually adopted for small engines. Good examples of such engines have been found by the Author to demand from 25 to 33 per cent less steam, or feed-water, than is above given.

The table on the next page gives what are considered by the Author as fair estimates of water and steam consumption for the best classes of engines in common use, when of moderate size and in good order.

It is considered usually advisable to assume a set of practically attainable conditions in average good practice, and to take

NON-CONDENSING ENGINES.

Steam Pressure.		Pounds per H. P. per Hour.—Ratio of Expansion.					
Atmospheres.	Lbs. per sq. in.	2	3	4	5	7	10
3	45	40	39	40	49	42	45
4	60	35	34	36	36	38	40
5	75	30	28	27	26	30	32
6	90	28	27	26	25	27	29
7	105	26	25	24	23	25	27
8	120	25	24	23	22	22	21
10	150	24	23	22	21	20	20

CONDENSING ENGINES.

4	30	30	28	28	30	35	40
5	45	28	27	27	26	28	32
4	60	27	26	25	24	25	27
5	75	26	25	25	23	22	24
6	90	26	24	24	22	21	20
8	120	25	23	23	22	21	20
10	150	25	23	22	21	20	19

the power so obtainable as the measure of the power of the boiler in commercial and engineering transactions. The unit generally assumed has been usually the weight of steam demanded per horse-power per hour by a fairly good steam-engine. This magnitude has been gradually decreasing from the earliest period of the history of the steam-engine. In the time of Watt, one cubic foot of water per hour was thought fair; at the middle of the present century, ten pounds of coal was a usual figure, and five pounds, commonly equivalent to about forty pounds of feed-water evaporated, was allowed the best engines. After the introduction of the modern forms of engine this last figure was reduced twenty-five per cent, and the most recent improvements have still further lessened the consumption of fuel and of steam. By general consent, the unit has now become thirty pounds of dry steam per horse-power per hour, which represents the performance of good non-condensing mill engines.

16. **General Schemes of Trial or Tests of Engines** are adopted by engineers which, while varying in detail, all closely

resemble each other in their main purposes, and are somewhat similar in methods. They commonly include boiler-trials as the only practicable and satisfactory means of ascertaining the quantity and quality of steam supplied, and the cost of power in steam, fuel, and money. They invariably involve the application of the indicator or the dynamometer, and, if complete, of both, for power measurements. When the question to be solved is simply the efficiency of engine, or of engine considered dynamically, of the engine as a machine, a comparison of the indicated with the dynamometric power gives the solution; but when the thermal efficiency and the efficiency in transformation of energy is to be measured, the measurement of the quantity of energy supplied in the form of heat, and hence a boiler-trial, must necessarily form a part of the operation. All general systems may therefore be said to involve the whole series of determinations of quantity already indicated; but the details have not yet been authoritatively prescribed in such manner as to fix a standard system or standard methods. The experience of the most experienced and distinguished practitioners is, however, gradually producing a tolerably well settled custom in the more important of the several operations involved. Some such methods and some general plans of test-trials will be later described.

17. Steam-Boiler Trials, apart from the engine-test, and made for the purpose of ascertaining the quantity and quality of steam made, its cost in fuel and in combustible matter contained therein, and the efficiency of the boiler and of its heating-surface, are now very generally made by a fairly well recognized system. In the United States and in Germany, particularly, such methods are now made to follow very generally the prescribed order of procedure devised and published by engineers of recognized standing. Such a standard system is that proposed by the committee of the American Society of Mechanical Engineers, and this standard will be that accepted in this work.*

* Transactions American Society of Mechanical Engineers, vol. vi., 1884. A Manual of Steam-Boilers, by R. H. Thurston (N. Y.: J. Wiley & Sons, 1888), chap. xiv. pp. 484-537.

In the operation of conducting any trial, we have, usually, a single, well-understood object to attain, and the engineer should accustom himself to carefully define that object in his own mind, and to as carefully describe that object in his instructions and regulations for the proposed trial. The whole operation can then be carried on with that point distinctly in view, and the proposed end can then be accomplished with maximum economy of time and labor, as well as with greatest exactness. The observations must be made by the engineer conducting the trial, or by his assistants, with this object clearly in mind, and each should have a well-defined part of the work assigned him, and should assume responsibility for that part, having a distinct understanding in regard to the extent of his responsibility, and a good idea of the extent and nature of the work done by his colleagues, and the relations of each part to his own. No observations should be permitted to be made by unauthorized persons for entrance upon the log; and no duties should be permitted to be delegated by one assistant to another, without consultation and distinct understanding with the engineer in charge. The aim of the observers is, in boiler trials, for example, to obtain an exact determination of the weight of fuel used, its proportion of combustible matter effective in developing heat, the exact weight of water evaporated under the known conditions of the trial, into steam, the determination of the character of that steam, and often the nature of the combustion and the composition of the furnace-gases. Each of these distinct objects requires the determination of certain well-defined quantities, and the observer to whom each set of observations is intrusted should, whenever possible, be made sufficiently well acquainted with the object to be attained, and the method to be pursued in reaching it, to be able to make his own readings with accuracy, and to work up the results correctly. It is only after he has acquired this knowledge that he can be expected to do his work without direct supervision, and with satisfactory precision. The trial should, wherever possible, be so conducted that any error that may occur in the record may be detected, checked, or, if advisable, removed, by some process of mutual verification of

related observations. It is in this direction that the use of graphical methods of record and automatic instruments has greatest value. We should lose no opportunity to introduce both.

18. Engine Trials may or may not include determination of boiler performance and efficiency; but if they are to be satisfactorily complete, measurements of the quantity and quality of the steam supplied are as essential as any other determinations of quantity. In some cases, only a comparison of the work done with its cost in fuel is called for; but in this case the total efficiency so obtained cannot be analyzed into the two factors, engine efficiency and boiler efficiency, and it is impossible to say to what extent engine or boiler is responsible for the final results obtained. The complete investigation of the action and performance of the engine, as a heat-engine and prime motor, must always include some method of obtaining a measure of the amount of heat-energy supplied to the machine; the proportion of that energy which reaches the engine in available form; the distribution and disposition of the available part; the extent to which it is converted into useful work and into wasted power; the amount in detail of the various wastes; the method as well as extent of wastes; and the ameliorating or exaggerating effect of any observable accidental or purposely produced variations of condition and of operation upon the wastes, the economies, and the several efficiencies of the engine.

It is thus important that ways should be found and methods practised, that will determine the quantity and quality—whether wet or dry—of the steam supplied, the pressures and volumes of every stage of transfer and of transformation, and the quantities of heat stored, conveyed, and utilized or wasted.

19. Engine and Boiler Tests are thus necessarily commonly conducted simultaneously in the settlement of important contracts, and essential data can only be thus secured. Where the quantities to be ascertained and measured are not likely to vary greatly with period of operation, a trial of a few hours' duration will answer all purposes. Gas-engines are often tested a single hour, and five hours is quite as long as is often de-

sirable. Steam-engine and boiler trials are seldom less than ten hours in length, often occupy a full day of twenty-four hours, sometimes last a week, unintermittedly, night and day, and it is even sometimes prescribed that the more important data shall be recorded for several months or for a year at a time. Ordinarily, a ten-hour trial is quite sufficient, if properly conducted.

20. **The Apparatus of Steam-boiler Trials** consists of tanks to receive and in which to weigh the feed-water; scales with which to effect these measurements and to weigh fuel and ashes; thermometers with which to determine the temperature of the water and steam, and pyrometers for the furnace-flue and chimney temperatures; and it is now usual to employ a calorimeter with which to determine the condition of the steam, and to measure the proportion of entrained water. Before the systems of boiler trial usually adopted are employed, it will be necessary to understand the methods of use and of calibration and of standardizing these various kinds of apparatus, the sources from which they may be obtained, and the best methods of their application to the securing of the needed data. It is usually thought best to *weigh* all water, rather than measure its volume. If measured, it should be carefully noted that its variation of density with temperature is considerable, and sufficient to introduce observable errors if a constant density is assumed.

The Apparatus of Engine-testing consists of steam-engine indicators, dynamometers, counters and gauges, and good timing instruments. The use of these instruments and the methods of test and correction are simple; and, with the exception of the indicator, none demands very extended notice. The indicator, however, is an instrument which must be made with the utmost possible care and skill, and the study and interpretation of its diagrams is a matter demanding some skill, knowledge, and experience. Considerable space will therefore be devoted to the description of this interesting and indispensable instrument, its uses and applications, to the study of the methods of interpretation of its record, and necessary measurement and computations.

CHAPTER II.

STEAM-BOILER TRIALS.*

21. The Object of a Trial of a Steam-boiler is to determine what is the quantity of steam that a boiler can supply under definitely prescribed conditions; what is the quality, as to moisture or dryness, of that steam; what is the amount of fuel demanded to produce that steam; what the character of the combustion, and the actual conditions of operation of the boiler when at work. The conditions prescribed for one trial may differ greatly from those of another trial, and such differences are often the essential matters to be studied. In any case it is assumed that the conditions under which the boiler is to be worked are to be definitely stated, and the engineer conducting the experiments is expected to ascertain all the facts which go to determine the performance of the boiler, and to state them with accuracy, conciseness, and completeness.

In the attempt to ascertain those facts the engineer meets with some difficulties, and finds it necessary to exercise the utmost care and skill. In conducting a steam-boiler trial the weight of the water supplied to the boiler must be determined; the weight of the fuel consumed must be obtained; the state of the steam made must be determined; and these quantities must all be noted at frequent intervals. It is also necessary to know whether the combustion is perfect or imperfect, and to what extent the conditions and facts noted are due to the boiler, and what to external conditions.

It has now come to be considered that the determination of power and economy of a steam-boiler demands all the care, skill, and perfection of method and of apparatus of any purely scientific investigation. It is desirable that all work of this kind shall be done in substantially the same way, in order that comparisons may be made.

* Mainly from the Author's "Manual of Steam Boilers," New York, published by John Wiley & Sons.

22. Tests of Value of Fuel are sometimes the sole object of a trial of a steam-boiler, the intent being to ascertain by actual experiment what quantity of water a fuel of unknown quality can evaporate in a boiler of which the general efficiency is fairly well established. In such cases the fuel is employed in the usual manner and the results compared with those obtained with fuels of known excellence. Thus, in a good type of boiler, having a good proportion of area of heating-surface to weight of fuel burned per hour, it may be found that a fuel of established reputation for uniform excellence will evaporate ten times its own weight of water "from and at" the boiling-point. The trial of a fuel of unknown quality may prove that this boiler will, under precisely similar conditions, evaporate an equal amount of water into steam, and yet the market price of the fuel may be considerably less than that of the other. The immediate result would be the substitution of the second for the first, should no counterbalancing disadvantages exist. In such cases the method of conducting the experiment is precisely the same as where the efficiency of the boiler is determined; but the object sought is quite a different one. This also commonly compels at least two trials, the one of the old and standard, the other of the new and uncertain fuel, and a comparison of boiler-efficiency as found in the two trials.

23. The Determination of the Value of a steam-boiler involves the measurement of its efficiency, independently of the nature of the fuel, and it is thus important that a standard system of measuring the effectiveness of the fuel should be settled upon, or that all variations of such effectiveness should be eliminated. The latter is commonly the course taken; and the determination of the efficiency of the boiler is based upon the measurement of the evaporation of water, under stated standard conditions, per unit weight of the combustible and burned portion of the fuel supplied during the trial.

But the power of the boiler is as important an element of its value as its efficiency, and a complete trial includes, usually, measurements of efficiency at both the rated and the maximum working power of the boiler as operated for its special purpose.

24. The Evaporative Power of Fuels depends upon

not only their chemical composition as fuels, but also to an important extent upon their structure and their physical condition in every aspect; on their greater or less purity, and the admixture of earths, moisture, or other foreign matters; the fitness of the furnace for their utilization; the air-supply; its quantity, temperature, and humidity; the proximity of chilling surfaces; the extent of the combustion-chamber in which the gases rising from the bed of coal or other combustible may be more or less completely consumed; and many other minor conditions, all of which tell, in a more or less important degree, upon their value and the efficiency of the system of heat-generation.

25. **Analyses of Fuels** are sometimes made, either as a check upon the results of the trial or in substitution for it. Should analysis show that a given fuel is rich in heat-producing elements, while trial fails to give the results that should have been obtained, and such as the use of other fuels in the same boilers indicates to be possible, it will at once appear that the fuel demands peculiar treatment, or some other arrangement of furnace. Should doubt exist which of a number of fuels of the same class is best, chemical analysis may give a quicker and cheaper answer to the question than a formal trial. It rarely happens, however, that any system is as satisfactory, in the end, as actual trial extending over so long a period as to eliminate uncertainties.

Methods of analysis differ somewhat. The following is a standard method of general treatment as prescribed by the Union of Engineers of Germany:*

In order to take a sample of the fuel, a shovelful from each barrow or wagon will be thrown into a box with a cover. The coal will be mixed up and spread in the form of a square upon a level floor, and then divided by two diagonals into four parts. Of these, two opposite parts will be taken away, the other two will be broken up small and mixed together. Another shovelful will then be thrown in, and the method continued until about 10 kilogrammes are in the box. This will then be

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* *American Engineer*, August, 1883.

closed and reserved for chemical analysis. For accurate experiments the halves which have been taken away should also be analyzed.

To determine the moisture in the coal, about 10 grammes from the above-named sample is to be heated for two hours to 105° or 110° C. The loss in weight shows the moisture in the coal. Coal which happens to have been wetted by rain or otherwise should not be used. The test should be applied to coal in the average state of moisture at which it is delivered from the pit mouth, and this state should, if necessary, be determined beforehand. The remainder of the sample, powdered and mixed thoroughly, serves to determine the ash, the carbon, the hydrogen, the nitrogen, and the sulphur. The heating-value of the coal is determined as follows: Suppose that it is found to contain c per cent of carbon, h per cent of hydrogen, s per cent of sulphur, o per cent of oxygen, and w per cent of water, then the theoretical heating-value is given by the formula of Dulong as follows:

(*a*). *Referred to Water at* 0° *Cent.*

$$8100c + 34320\left(h - \frac{o}{8}\right) + 2500s.$$

(*b*). *Referred to Water at* 100° *Cent.*

$$8100c + 34200\left(h - \frac{o}{8}\right) + 2500s - 636.5\,(9h + w.)$$

To determine the quantity of air required for burning coal we have the following: One kilogramme of coal requires to burn it,

$$\frac{2.667c + 8h + s - o}{100 \times 1.43} \text{ cu. metres of oxygen ; or,}$$

$$\frac{2.667c + 8h + s - o}{21 \times 1.43} \text{ cu. metres of air containing 21 per ct. of oxygen.}$$

The analyses should be made with care, by a skilled and experienced chemist, if any important question is to be settled.

26. Economy of Fuel is nearly synonymous with efficiency of boiler, as a matter of engineering simply; but when the finance of the case is studied, it is often found, from that

point of view, a very different mattter. It is perfectly possible to adopt so great a proportion of heating-surface, so large a boiler, that the gain in fuel saved, as compared with boilers of similar type and usual proportions, may be more than offset by the increased charges on account of enlargement of boiler. The efficiency of boiler, in the ordinary sense in which that term is used, is, however, a measure of economy. The variation of efficiency and of economy in fuel consumption is a function of the proportion of area and of heating-surface to fuel burned, and the object of a boiler-trial is to ascertain these relations with precision. An understanding should be had before the trial in regard to the kind of fuel to be used; where no reason of controlling importance exists to the contrary, the best obtainable coal should be selected, for the reason that a boiler can be better judged, and the results of its trial may be more satisfactorily compared with similar trials of other boilers, when the very best work of which it is capable is done by it. The differences between separate lots of the best coals are less than the differences between separate lots of inferior fuels, and the comparison is thus less difficult where the former are used.

27. **The Relative Values of Boilers** depend not only on their efficiencies, but also on their capacities for furnishing steam, and on various other qualities and attributes: as their greater or less complication in structure; their safety and durability; their volume, weight, and cost. The boiler-trial only settles questions relating to their efficiency and capacity, and their real relations of value, only just so far as those elements enter the problem. These are usually, however, the main factors, and their measurement by a test-trial gives the means of deciding, in nearly all cases, every question likely to present itself in the use of the apparatus.

28. **Variations of Efficiency** occur with variations in grate-area, in rate of combustion and in kind of fuel. In any given boiler, within a wide range of which the limits are usually far outside of practical conditions, the greater the quantity of fuel burned the less the amount of steam made per unit weight of that fuel, the smaller the quantity of fuel,

burned under proper conditions, in the boiler, the higher the efficiency; and it has been seen in an earlier chapter, that the gain in efficiency, with increasing proportion of heating to grate surface or to fuel burned, is less and less as this increase goes on. By enlarging or reducing the grate, or by increasing or diminishing the draught and air-supply, and during a succession of trials, noting the method of variation of efficiency and of capacity for making steam, the law of such variations may be established, and the best arrangement, all things considered, may be determined.

29. Variations of Proportions in different boilers, otherwise similar, have been seen to be capable of expression by a very simple algebraic expresssion on which all theories of efficiency are based. But in some cases this law is not found to be precisely applicable, and only test-trials of boilers so differing can be relied upon to give correct relations. The general relations already stated invariably hold; but it often happens that a steam-boiler exhibits peculiarities which make that exact statement inapplicable. It is not uncommon not only to compare actual performance, as shown by trial, with the results indicated by the theory, but also to alter the ratio of heating to grate surface by bricking over more or less of the grate, and by this or other expedients so varying that ratio in successive trials as to obtain an empirical and approximately exact expression for the law of variation of efficiency for the particular case in hand.

30. Combined Power and Efficiency distinguish the best types of boiler. That which, at a given cost, exhibits highest steam-producing power combined with greatest efficiency, is the best boiler. These qualities, however, are not usually compatible, and increased steam-production from any boiler is commonly attended with a decrease in efficiency; and as the one or the other of these qualities is the more important, the combination which will give best total result will vary. In no two cases will the same combination be equally desirable. Every boiler must be tested for both before it can be said whether it is satisfactorily adapted to its place and work.

31. The Apparatus and Methods of test-trials should be

prescribed in the preliminary arrangements for every trial, and if possible should be in exact accordance with some accepted standard rules. The apparatus consists of scales and tanks for measurement of weights of coal and of water; gauges to give the pressure of steam; thermometers of great accuracy to determine the temperatures of water, steam, and flue-gases; and calorimeters to determine the quality of the steam and the extent of superheating, or the percentage of moisture entrained by it.

The establishment of the correctness of this apparatus is the first of the preliminaries to their use. The standardization of the instruments is a matter of supreme importance, since upon their accuracy the whole work of the engineer is dependent. It is also a work demanding, in most cases, unusual skill and care, and, to be satisfactory, must generally be performed either at the manufacturer's, or at the office of the engineer conducting the trial. The scales can usually be standardized by the official sealer of weights and measures, and sealed by him; the water-meters, if used, can be readily tested by the use of the scales so sealed; the thermometers are, as a rule, best tested by their makers, and should be sent to the maker for test immediately before and directly after the test. The engineer often has a carefully preserved standard with which they may be compared in his own office. The same remarks apply to the examination of the gauges used, which should be standardized both before and after their use. The apparatus used in connection with the calorimeter, in the determination of the quality of the steam made, demand exceptional care in this process. Where it is unavoidable, the use of coarsely graduated thermometers and roughly constructed scales may be permitted, but only then when a very large number of observations are taken, and an average thus obtained which may befairly expected to fall within reasonable limits of error.

The method of starting and of stopping the trial is a very important matter, and one upon which engineers of experience and acknowledged authority are not in complete accord. The principles to be adhered to in this matter, as in every other detail of the operation of testing a boiler, are easily specified,

but they are not always as easy of practice. All conditions should be as exactly the same at the beginning and at the end of the test as they can possibly be made. The period of the trial and the times of stopping and of starting should be capable of being exactly fixed, and the method of test should be such as should permit of the commencement and the end occurring at these exactly defined times, or, as an alternative, they should be such that the work done by the boiler during the less precisely determinable time of beginning and ending of the trial should be as nearly as possible *nil*, so that a slight error as to time may not appreciably affect the results.

During the trial, provision should be made for the preservation of the utmost possible uniformity of working conditions throughout the whole period of the trial. Every irregularity gives rise to more or less loss of efficiency, and to uncertainty in regard to the correctness of the reported figures. The nearer the working of the boiler is kept to the final average for the trial, the better.

Uniformity of operation and maximum efficiency are best attainable during a trial when a system of record is adopted which allows of that regularity being shown at all times; and records in proper form are the best possible security against error of observation. Graphical methods should be adopted wherever practicable. Such methods of record exhibit most satisfactorily the accordance with or the deviation from the uniformity of operation considered so desirable on the score of efficiency and accuracy.

32. **Standard Test-trials** are made under established systems, and in accordance with codes of regulations which are accepted as representing a satisfactory system of procedure. In such cases the first step is to settle upon a standard of measurement and comparison that may be accepted by all who may be interested in the result. The standard nominal horse-power has already been described as now accepted by the best authorities.

The Committee of Judges of the Centennial Exhibition, to whom the trials of competing boilers at that exhibition were intrusted, adopted the unit, 30 *pounds of water evaporated into*

dry steam per hour from feed-water at 100° *Fahrenheit, and under a pressure of seventy pounds per square inch above the atmosphere*, these conditions being considered to represent fairly average practice The quantity of heat demanded to evaporate a pound of water under these conditions is 1110.2 British thermal units, or 1.1496 "units of evaporation." The unit of power proposed is thus equivalent to the development of 33,305 heat-units per hour, or 34.488 units of evaporation. The "unit of evaporation" is taken as a certain weight—preferably unity of water, evaporated "from and at" the boiling-point under atmospheric pressure. The now-accepted unit of boiler-power, in the code constructed for the American Society of Mechanical Engineers,* is the equivalent of the Centennial Standard, and in all standard trials the commercial horse-power is taken as *an evaporation of* 30 *pounds of water per hour from a feed-water temperature of* 100° *Fahr. into steam at* 70 *pounds gauge-pressure*, which is equal to 34½ units of evaporation, that is, to 34½ pounds of water evaporated from a feed-water temperature of 212° Fahr. into steam at the same temperature. This standard is taken as the equivalent of 33,305 thermal units per hour.†

A boiler rated at any stated horse-power should be capable of developing that power with easy firing, moderate draught and ordinary fuel, while exhibiting good economy; and the boiler should be capable of developing one half or one third more than its rated power to meet emergencies at times when maximum economy is not the most important object to be attained.

33. Instructions and Rules governing the standard system of boiler-trial, prepared by a committee of the American Society of Mechanical Engineers, may be taken as a good illustration of such regulations as, in one form or another, have

* Transactions, vol. vi, 1884.

† An evaporation of 30 pounds of water from 100° F. into steam at 70 pounds pressure is equal to an evaporation of 34.488 pounds from and at 212°, and an evaporation of 34½ pounds from and at 212° F. is equal to 30.010 pounds from 100° F., into steam at 70 pounds pressure.

The "unit of evaporation" being taken equal to 965.7 thermal units, the commercial horse-power is $34.488 \times 965.7 = 33.305$ thermal units

been customarily agreed upon by engineers conducting such work. They are as follows:

PRELIMINARIES TO A TEST.

I. *In preparing for* and conducting trials of steam-boilers, the specific object of the proposed trial should be clearly defined and steadily kept in view.

II. *Measure and record the dimensions*, position, etc., of grate and heating surfaces, flues and chimneys, proportion of air-space in the grate-surface, kind of draught, natural or forced.

III. *Put the Boiler in good condition.*—Have heating-surface clean inside and out, grate-bars and sides of furnace free from clinkers, dust and ashes removed from back connections, leaks in masonry stopped, and all obstructions to draught removed. See that the damper will open to full extent, and that it may be closed when desired. Test for leaks in masonry by firing a little smoky fuel and immediately closing damper. The smoke will then escape through the leaks.

IV. *Have an understanding with the parties* in whose interest the test is to be made as to the character of the coal to be used. The coal must be dry, or, if wet, a sample must be dried carefully and a determination of the amount of moisture in the coal made, and the calculation of the results of the test corrected accordingly.

Wherever possible, the test should be made with standard coal of a known quality. For that portion of the country east of the Alleghany Mountains good anthracite egg coal or Cumberland semi-bituminous coal may be taken as the standard for making tests. West of the Alleghany Mountains and east of the Missouri River, Pittsburg lump coal may be used.*

V. *In all important tests* a sample of coal should be selected for chemical analysis.

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* These coals are selected because they are almost the only coals which contain the essentials of excellence of quality, adaptability to various kinds of furnaces, grates, boilers, and methods of firing, and wide distribution and general accessibility in the markets.

VI. *Establish the correctness of all apparatus* used in the test for weighing and measuring. These are:

1. Scales for weighing coal, ashes, and water.

2. Tanks, or water-meters for measuring water. Water-meters, as a rule, should only be used as a check on other measurements. For accurate work, the water should be weighed or measured in a tank.

3. Thermometers and pyrometers for taking temperatures of air, steam, feed-water, waste gases, etc.

4. Pressure-gauges, draught-gauges, etc.

VII. *Before beginning a test*, the boiler and chimney should be thoroughly heated to their usual working temperature. If the boiler is new, it should be in continuous use at least a week before testing, so as to dry the mortar thoroughly and heat the walls.

VIII. Before beginning a test, the boiler and connections should be free from leaks, and all water-connections, including blow and extra-feed pipes, should be disconnected or stopped with blank flanges, except the particular pipe through which water is to be fed to the boiler during the trial. In locations where the reliability of the power is so important that an extra feed-pipe must be kept in position, and in general when for any other reason water-pipes other than the feed-pipes cannot be disconnected, such pipes may be drilled so as to leave openings in their lower sides, which should be kept open throughout the test as a means of detecting leaks, or accidental or unauthorized opening of valves. During the test the blow-off pipe should remain exposed.

If an injector is used, it must receive steam directly from the boiler being tested, and not from a steam-pipe, or from any other boiler.

See that the steam-pipe is so arranged that water of condensation cannot run back into the boiler. If the steam-pipe has such an inclination that the water of condensation from any portion of the steam-pipe system may run back into the boiler, it must be trapped so as to prevent this water getting into the boiler without being measured.

STARTING AND STOPPING A TEST.

A test should last at least ten hours of continuous running, and twenty-four hours whenever practicable. The conditions of the boiler and furnace in all respects should be, as nearly as possible, the same at the end as at the beginning of the test. The steam-pressure should be the same, the water-level the same, the fire upon the grates should be the same in quantity and condition, and the walls, flues, etc., should be of the same temperature. To secure as near an approximation to exact uniformity as possible in conditions of the fire and in temperatures of the walls and flues, the following method of starting and stopping a test should be adopted:

X. *Standard Method.*—Steam being raised to the working pressure, remove rapidly all the fire from the grate, close the damper, clean the ash-pit, and as quickly as possible start a new fire with weighed wood and coal, noting the time of starting the test and the height of the water-level while the water is in a quiescent state, just before lighting the fire.

At the end of the test, remove the whole fire, clean the grates and ash-pit, and note the water-level when the water is in a quiescent state; record the time of hauling the fire as the end of the test. The water-level should be as nearly as possible the same as at the beginning of the test. If it is not the same, a correction should be made by computation, and not by operating pump after test is completed. It will generally be necessary to regulate the discharge of steam from the boiler tested by means of the stop-valve for a time while fires are being hauled at the beginning and at the end of the test, in order to keep the steam-pressure in the boiler at those times up to the average during the test.

XI. *Alternate Method.*—Instead of the Standard Method above described, the following may be employed where local conditions render it necessary:

At the regular time for slicing and cleaning fires have them burned rather low, as is usual before cleaning, and then thoroughly cleaned: note the amount of coal left on the grate as nearly as it can be estimated; note the pressure of

steam and the height of the water-level—which should be at the medium height to be carried throughout the test—at the same time, and note this time as the time of starting the test. Fresh coal, which has been weighed, should now be fired. The ash-pits should be thoroughly cleaned at once after starting. Before the end of the test the fires should be burned low, just as before the start, and the fires cleaned in such a manner as to leave the same amount of fire, and in the same condition, on the grates as at the start. The water-level and steam-pressure should be brought to the same point as at the start, and the time of the ending of the test should be noted just before fresh coal is fired.

DURING THE TEST.

XII. *Keep the Conditions Uniform.*—The boiler should be run continuously, without stopping for meal-times or for rise or fall of pressure of steam due to change of demand for steam. The draught being adjusted to the rate of evaporation or combustion desired before the test is begun, it should be retained constant during the test by means of the damper.

If the boiler is not connected to the same steam-pipe with other boilers, an extra outlet for steam with valve in same should be provided, so that in case the pressure should rise to that at which the safety-valve is set, it may be reduced to the desired point by opening the extra outlet, without checking the fires.

If the boiler is connected to a main steam-pipe with other boilers, the safety-valve on the boiler being tested should be set a few pounds higher than those of the other boilers, so that in case of a rise in pressure the other boilers may blow off, and the pressure be reduced by closing their dampers, allowing the damper of the boiler being tested to remain open, and firing as usual.

All the conditions should be kept as nearly uniform as possible, such as force of draught, pressure of steam, and height of water. The time of cleaning the fires will depend upon the character of the fuel, the rapidity of combustion, and the kind of grates. When very good coal is used, and the combustion not too rapid, a ten-hour test may be run without any cleaning

of the grates, other than just before the beginning and just before the end of the test. But in case the grates have to be cleaned during the test, the intervals between one cleaning and another should be uniform.

XIII. *Keeping the Records.*—The coal should be weighed and delivered to the firemen in equal portions, each sufficient for about one hour's run, and a fresh portion should not be delivered until the previous one has all been fired. The time required to consume each portion should be noted, the time being recorded at the instant of firing the first of each new portion. It is desirable that at the same time the amount of water fed into the boiler should be accurately noted and recorded, including the height of the water in the boiler, and the average pressure of steam and temperature of feed during the time. By thus recording the amount of water evaporated by successive portions of coal, the record of the test may be divided into several divisions, if desired, at the end of the test, to discover the degree of uniformity of combustion, evaporation, and economy at different stages of the test.

XIV. *Priming Tests.*—In all tests in which accuracy of results is important, calorimeter tests should be made of the percentage of moisture in the steam, or of the degree of superheating. At least ten such tests should be made during the trial of the boiler, or so many as to reduce the probable average error to less than one per cent, and the final records of the boiler test corrected according to the average results of the calorimeter tests.

On account of the difficulty of securing accuracy in these tests the greatest care should be taken in the measurements of weights and temperatures. The thermometers should be accurate to within a tenth of a degree, and the scales on which the water is weighed to within one hundredth of a pound.

ANALYSES OF GASES.—MEASUREMENT OF AIR-SUPPLY, ETC.

XV. In tests for purposes of scientific research, in which the determination of all the variables entering into the test is desired, certain observations should be made which are in general not necessary in tests for commercial purposes. These are the measurement of the air-supply, the determination of its con-

tained moisture, the measurement and analysis of the flue-gases, the determination of the amount of heat lost by radiation, of the amount of infiltration of air through the setting, the direct determination by calorimeter experiments of the absolute heating value of the fuel, and (by condensation of all the steam made by the boiler) of the total heat imparted to the water.

The analysis of the flue-gases is an especially valuable method of determining the relative value of different methods of firing, or of different kinds of furnaces. In making these analyses great care should be taken to procure average samples, since the composition is apt to vary at different points of the flue, and the analyses should be intrusted only to a thoroughly competent chemist, who is provided with complete and accurate apparatus.

As the determination of the other variables mentioned above are not likely to be undertaken except by engineers of high scientific attainments, and as apparatus for making them is likely to be improved in the course of scientific research, it is not deemed advisable to include in this code any specific directions for making them.

RECORD OF THE TEST.

XVI. A "log" of the test should be kept on properly prepared blanks, containing headings as follows:

Time	Pressures.			Temperatures.					Fuel.		Feed-water.	
	Barometer	Steam gauge	Draught gauge	External Air	Boiler-room	Flue	Feed water.	Steam.	Time.	Pounds.	Time	Pounds or c ft

REPORTING THE TRIAL.

XVII. The final results should be recorded upon a properly prepared blank, and should include as many of the following items as are adapted for the specific object for which the trial is made. The items marked with a * may be omitted for ordinary trials, but are desirable for comparison with similar data from other sources.

Results of the trials of a..............................

Boiler at.......................................

To determine......................................

1. Date of trial..............................			
2. Duration of trial...........................	hours.		
DIMENSIONS AND PROPORTIONS.			
Leave space for complete description. See Appendix XXIII.			
3. Grate-surface... wide....long....Area....	sq. ft.		
4. Water-heating surface.....................	sq. ft.		
5. Superheating-surface......................	sq. ft.		
6. Ratio of water heating surface to grate-surface..............................			
AVERAGE PRESSURES.			
7. Steam-pressure in boiler, by gauge........	lbs.		
*8. Absolute steam pressure	lbs.		
*9. Atmospheric pressure, per barometer.......	in.		
10. Force of draught in inches of water........	in.		
AVERAGE TEMPERATURES.			
*11. Of external air...........................	deg.		
*12. Of fire-room..............................	deg.		
*13. Of steam..................................	deg.		
14. Of escaping gases.........................	deg.		
15. Of feed-water.............................	deg.		
FUEL.			
16. Total amount of coal consumed †..........	lbs.		
17. Moisture in coal..........................	per cent.		
18. Dry coal consumed.........................	lbs.		
19. Total refuse. dry......pounds =..........	per cent.		
20. Total combustible (dry weight of coal, Item 18, less refuse, Item 19)	lbs.		
*21. Dry coal consumed per hour...............	lbs.		
*22. Combustible consumed per hour............	lbs.		

* See reference in paragraph preceding table.

† Including equivalent of wood used in lighting fire. 1 pound of wood equals 0.4 pound coal. Not including unburnt coal withdrawn from fire at end of test.

RESULTS OF CALORIMETRIC TESTS.			
23 Quality of steam, dry steam being taken as unity			
24. Percentage of moisture in steam............	per cent.		
25 Number of degrees superheated...........	deg.		
WATER.			
26. Total weight of water pumped into boiler and apparently evaporated *............	lbs.		
27. Water actually evaporated, corrected for quality of steam †	lbs.		
28. Equivalent water evaporated into dry steam from and at 212° F †....................	lbs.		
*29. Equivalent total heat derived from fuel in British thermal units †...................	B. T. U.		
30. Equivalent water evaporated into dry steam from and at 212° F. per hour............	lbs.		
ECONOMIC EVAPORATION.			
31. Water actually evaporated per pound of dry coal, from actual pressure and temperature †	lbs.		
32. Equivalent water evaporated per pound of dry coal from and at 212° F †..........	lbs.		
33. Equivalent water evaporated per pound of combustible from and at 212° F †.	lbs.		

* Corrected for inequality of water-level and of steam-pressure at beginning and end of test.

† The following shows how some of the items in the above table are derived from others:

Item 27 = Item 26 × Item 23

Item 28 = Item 27 × Factor of evaporation.

Factor of evaporation $= \frac{H - h}{965.7}$, H and h being respectively the total heat-units in steam of the average observed pressure and in water of the average observed temperature of feed, as obtained from tables of the properties of steam and water.

Item 29 = Item 27 × $(H - h)$.

Item 31 = Item 27 ÷ Item 18.

Item 32 = Item 28 ÷ Item 18 or = Item 31 × Factor of evaporation.

Item 33 = Item 28 ÷ Item 20 or = Item 32 ÷ (per cent 100 − Item 19).

Items 36 to 38. First term = Item 20 × $\frac{6}{5}$

Items 40 to 42. First term = Item 39 × 0.8698.

Item 43 = Item 29 × 0.00003 or = $\frac{\text{Item 30}}{34\frac{1}{2}}$.

Item 45 = $\frac{\text{Difference of Items 43 and 44}}{\text{Item 44}}$.

COMMERCIAL EVAPORATION			
34. Equivalent water evaporated per pound of dry coal with one sixth refuse at 70 pounds gauge pressure, from temperature of 100° F. = Item 33 multiplied by 0.7249	lbs.		
RATE OF COMBUSTION.			
35. Dry coal actually burned per square foot of grate-surface per hour.......	lbs.		
*36. Consumption of dry coal per hour. Coal assumed with one sixth refuse.† — Per sq. ft. of grate-surface...	lbs.		
*37. Consumption of dry coal per hour. Coal assumed with one sixth refuse.† — Per sq. ft. of water-heating surface ..	lbs.		
*38. Consumption of dry coal per hour. Coal assumed with one sixth refuse.† — Per sq. ft. of least area for draught...	lbs.		
RATE OF EVAPORATION			
39. Water evaporated from and at 212° F. per square foot of heating-surface per hour...	lbs.		
*40. Water evaporated per hour from temperature of 100° F into steam of 70 pounds gauge-pressure † — Per sq. ft. of grate-surface.........	lbs.		
*41. Water evaporated per hour from temperature of 100° F into steam of 70 pounds gauge-pressure † — Per sq. ft. of water-heating surface.	lbs.		
*42. Water evaporated per hour from temperature of 100° F into steam of 70 pounds gauge-pressure † — Per sq. ft. of least area for draught.	lbs.		
COMMERCIAL HORSE POWER.			
43. On basis of thirty pounds of water per hour evaporated from temperature of 100° F into steam of 70 pounds gauge pressure, (= 34½ lbs. from and at 212°) †	H. P.		
44. Horse power, builders' rating, at.... square feet per horse power............	H. P.		
45. Per cent developed above, or below, rating †....................................	Per cent.		

34. Precautions are to be taken in every possible way to prevent and avoid irregularities in the conduct of the trial and errors of observation.*

In preparing for and conducting trials of steam-boilers the specific object of the proposed trial should be clearly defined and steadily kept in view, and as suggested by Mr. Hoadley—

(1) If it be to determine the efficiency of a given style of boiler or of boiler-setting under normal conditions, the boiler brickwork, grates, dampers, flues, pipes, in short, the whole apparatus, should be carefully examined and accurately described,

* The appendix to the report above quoted should be read in this connection.

and any variation from a normal condition should be remedied, if possible, and if irremediable, clearly described and pointed out.

(2) If it be to ascertain the condition of a given boiler or set of boilers with a view to the improvement of whatever may be faulty, the conditions actually existing should be accurately observed and clearly described.

(3) If the object be to determine the relative value of two or more kinds of coal, or the actual value of any kind, exact equality of conditions should be maintained if possible, or, where that is not practicable, all variations should be duly allowed for.

(4) Only one variable should be allowed to enter into the problem: or, since the entire exclusion of disturbing variations cannot usually be effected, they should be kept as closely as possible within narrow limits, and allowed for with all possible accuracy.

Blanks should be provided in advance, in which to enter all data observed during the test. The preceding instructions contain the form used in presenting the general results. Records should be, as far as possible, made in a standard form, in order that all may be comparable.

The observations must be made by the engineer conducting the trial, or by his assistants, with this object distinctly in mind; and each should have a well-defined part of the work assigned him, and should assume responsibility for that part, having a distinct understanding in regard to the extent of his responsibility, and a good idea of the extent and nature of the work done by his colleagues, and the relations of each part to his own. No observations should be permitted to be made by unauthorized persons for entrance upon the log; and no duties should be permitted to be delegated by one assistant to another, without consultation and distinct understanding with the engineer in charge. The trial should, wherever possible, be so conducted that any error that may occur in the record may be detected, checked, or, if advisable, removed, by some process of mutual verification of related observations. It is in this direction that the use of graphical methods of record and automatic instruments have greatest value.

Several methods of weighing fuel have been found very satisfactory, but it should be an essential feature that the weights shall be made by one observer and checked by another, at as distant a point as is convenient. The weighing of the fuel by one observer at the point of storage, and the record at that point of times of delivery, as well as of weights of each lot, and the tallying of the number and record of the time of receipt at the furnace-door, will be usually found a safe system. The failure to record any one weight leads to similar error, and can only be certainly prevented by an effective method of double observation and check.

The same remarks apply, to a considerable extent, to the weighing of the water fed to the boiler. A careful arrangement of weighing apparatus, a double set of observations, where possible, and thus safe checks on the figures obtained, are essential to certainty of results. With good observers at the tank, and with small demand for water, a single tank can be used; but two are preferable in all cases, and three should be used if the work demands very large amounts of feed-water, as at trials of very large boilers, or of "batteries." The more uniform the water-supply, as well as the more steady the firing, the less the liability to mistake in making the record.

The two blanks which follow were prepared by the Author for use in laboratory as well as professional work.

Such blanks are always desirable, and are sometimes even more elaborate than those here illustrated.* For ordinary purposes, a less complete tabulation may be sufficient; while, for special purposes and for scientific investigations and the work of research, considerable more elaboration may be found desirable, or even necessary.

Graphical methods of representation of the data and conditions and of the progress of the trial, as elsewhere illustrated, are also often found exceedingly useful and convenient.

* *Vide* "Stevens Indicator," Nov. '89.

TABLE I.

LOG OF TRIAL BY MECHANICAL LABORATORY, DEPARTMENT OF ENGINEERING.

Test made at ____________________

____

on ____________________

$$x = \frac{U - wh}{H - h}$$

$$\frac{\frac{U}{w} + t - T}{0.48} = \text{Degrees of Superheating.}$$

Time.	Pressures.			Temperatures.					Weights.			Remarks.
	Barometer.	Steam-gauge	Draught-gauge.	External Air.	Boiler-room.	Fuel.	Feed-water.	Steam.	Fuel.	Feed-water.		
										Per Metre.	Per Tank.	

PRIMING TESTS.

No.	Time.	Steam-pressures.	Calorimeter.					Heat-units per Pound from Boiler		Heat transferred to Calorimeter. $W \times R - U$	Heat from Steam $H = T - t'$	Heat from Water $h = t - t'$	Steam run into Calorimeter. x	Percentage of Priming. y	Superheating.		Remarks.
			Weights.		Temperature.												
			Condensing Water. W	Wet Steam w	Initial t''	Final t'	Range. $R = t' - t''$	Water t	Steam. T						Degrees.	Heat-units.	

TABLE II.

AVERAGE AND TOTAL RESULTS OF TRIAL, MECHANICAL LABORATORY, DEPARTMENT OF ENGINEERING.

Trial made at__________________ Fuel__________________

on__________________ Composition__________________

Number of Trial.	Date of Trial.	Length of Trial.	Areas.				Ratio of Grate to Heating-surface.	Average Temperatures.				Average Pressures.			Consumption of Fuel.			Ashes.		Total Combustible.	Remarks.
			Grate	Heating surface.	Super heating surface	Least Cross section of Flues.		Boiler-room	External Air	Entrance to Chimney	Feed-water.	Barometer	Steam-gauge	Draught gauge	Total	Per square foot of Grate per hour	Per sq ft of Heating-surface per hour	Total	Proportion of Total Fuel		
		Hours.	sq ft.	sq. ft.	sq ft	sq ft.		Fahr.	Fahr	Fahr.	Fahr	ins.	lbs	ins	lbs.	lbs.	lbs.	lbs.	per ct.	lbs.	

Apparent Evaporation.									Real Evaporation.											
Per Pound of Fuel.			Per Pound of Combustible			Per Square Foot of Heating-surface, per Hour.			Per Pound of Fuel.			Per Pound of Combustible.			Per Square Foot of Heating-surface, per Hour			Square feet of Heating-surface required to Evaporate one Cubic Foot of Water.		
From actual temperature of feed-water and at actual steam pressure	Equivalent from and at 212° F	Equivalent from 212° F and at actual steam pressure	From actual temperature of feed water and at actual steam pressure	Equivalent from and at 212° F	Equivalent from 212° F and at actual steam-pressure	From actual temperature of feed-water and at actual steam-pressure	Equivalent from and at 212° F.	Equivalent from 212° F and at actual steam-pressure	From actual temperature of feed-water and at actual steam-pressure	Equivalent from and at 212° F	Equivalent from 212° F and at actual steam-pressure	From actual temperature of feed-water and at actual steam-pressure	Equivalent from and at 212° F	Equivalent from 212° F and at actual steam pressure	From actual temperature of feed water and at actual steam pressure	Equivalent from and at 212° F	Equivalent from 212° F and at actual steam pressure	From actual temperature of feed-water and at actual steam-pressure	Equivalent from and at 212° F	Equivalent from 212° F and at actual steam-pressure.
lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs	lbs.	lbs.	lbs	sq ft	sq ft	sq ft.

TABLE II.—*Continued.*

Total Water fed to Boiler.			Average Priming	Total Water Primed.	Water Evaporated into Dry Steam.			Remarks.
From actual temperature of feed-water and at actual steam-pressure.	Equivalent from and at 212° F.	Equivalent from 212° F. and at actual steam-pressure.			From actual temperature of feed-water and at actual steam-pressure.	Equivalent from and at 212° F.	Equivalent from 212° F. and at actual steam-pressure.	
lbs.	lbs.	lbs.	per cent.	lbs.	lbs.	lbs.	lbs.	

Average Amount of Superheating.	Evaporation from and at 212° F., Equivalent to Total Heat-units derived from Fuel.			Efficiency.			Values of A and B in $F = A\sqrt{H} + B$.		Horse-power.		Remarks.
	Per Pound of Fuel.	Per Pound of Combustible.	Per sq. ft. of Heating-surface per hour.	Experimental.	Estimated.	$R = \frac{\text{Experimental.}}{\text{Estimated.}}$	A	B	Rated.	Actual.	
Fahr.	lbs.	lbs.	lbs.	per cent.	per cent.						

The following is a good form of simple, but usually sufficient, boiler-room log, for everyday purposes: *

........Mo........188 .	Hours.												Total.	Average.
	6	7	8	9	10	11	12	1	2	3	4	5		
B. Press by gauge														
Fuel Fired, lbs.														
Ashes and uncons														
Combustible, lbs.														
Uptake temp.....														
Feed temp														
. .. .														
Turns per min...														
Throttle open														
Av Cut-off														
Av Vacuum, in.														
Hot-well temp														
Injection temp														
" lbs														
Indicated H P..... ...														
... .														

Total hours run ....
Kind of coal...............................
Cost coal per 2,240 lbs.....................
Oil, gals . ...
Kind of Oil.... ....
Cost of oil per gal ....
Waste, lbs.................................

Steam, dry or wet ... .
Lbs. water per lb. coal .. . .
Lbs. water per lb. combustible .. ..
Lbs. " " coal from and at 212°..
Lbs. " " comb. from and at 212°.
Remarks:

35. The Heating Power of any Fuel is determined by calculating its *total heat of combustion.* This quantity is the sum of the amounts of heat generated by the combustion of the unoxidized carbon and hydrogen contained in the fuel, less the heat required in the evaporation and volatilization of constituents which become gaseous at the temperature resulting from the combustion of the first-named elements. It is measured in "thermal units."

A *thermal unit* is the quantity of heat necessary to raise a unit weight of water, at temperature of maximum density, one degree of temperature. The British thermal unit is the quantity of heat required to raise a pound of water from the temperature 39°.1 to 40°.1 Fahr. The metric unit or calorie is the

* See The Practical World, Dec. 1888, p. 1.

quantity of heat required to raise one kilogramme of water (2.2046215 pounds) from 3°.94 to 4°.94 Centigrade.

One metric or Centigrade unit is equal to 3.96832 British units, and a British unit is equal to 0.251996 metric unit.

An approximate estimate of the number of thermal units developed by the combustion of a pound or kilogramme of any dry fuel, of which the chemical composition is known, may be obtained by the use of the following formula:

$$\left.\begin{aligned} h &= 14{,}500C + 62{,}000\left(H - \frac{O}{8}\right), \\ h' &= 8{,}080C + 34{,}460\left(H - \frac{O}{8}\right), \end{aligned}\right\}$$

where h is the number of British thermal units representing the total heat of combustion of one pound of the fuel; h' is the number of metric units per kilogramme of fuel; C represents the percentage of carbon; H that of hydrogen; and O that of oxygen.

Thus an anthracite coal has been found to have the following composition:

COMPOSITION OF ANTHRACITE COAL.

	Per cent.
Carbon	81.34
Hydrogen *uncombined*	3.45
Hydrogen *in combination*	0.74
Oxygen and Nitrogen	5.89
Sulphur	0.64
Water	2.00
Ash	5.94
Total	100.00

One pound or kilogramme of coal, of which the above is an analysis, can evaporate theoretically 14.4 pounds or kilogrammes of water from and at 100° Centigrade, or 212° Fahr.

MM. Scheurer, Kestner, and Meunier have adopted the common formula as first proposed by Dulong, but would omit

all account of oxygen, thus reducing, as is claimed, the average error of the formula from about 12 per cent or more to 8 or 10. M. Cornut would separate the fixed from the volatile carbon, and would give the latter about one third more credit for heating power than the former; closer approximations are thus made than by the other methods.

Mr. G. H. Babcock gives the following tables as representative of familiar practice:

Kind of Combustible.	Air Required.	Temperature of Combustion.				Theoretical Value.		Highest Attainable Value under Boiler.	
	In lbs. per pound of combustible.	With Theoretical Supply of Air.	With 1½ times the theoretical supply of air	With twice the theoretical supply of air.	With three times the theoretical supply of air	In lbs. of water raised 1° per lb of combustible.	In lbs. of water evaporated from and at 212°, with 1 lb. combustible.	With chimney draught.	With blast, theoretical supply of air at 60°, gas 320°.
Hydrogen.	36.00	5,750	3,860	2,860	1,940	62,032	64.20		
Petroleum.	15.43	5,050	3,515	2,710	1,850	21,000	21.74	18.55	19.9
Carbon—									
Charcoal, Coke, Anthracite C'l	12.13	4,580	3,215	2,440	1,650	14,500	15.00	13.90	14.14
Coal—									
Cumberland	12.06	4,900	3,360	2,550	1,730	15,370	15.90	14.28	15.06
Coking bituminous	11.73	5,140	3,590	2,680	1,810	15,837	16.00	14.45	15.19
Cannel	11.80	4,850	3,330	2,540	1,720	15,080	15.60	14.01	14.76
Lignite	9.30	4,600	3,210	2,490	1,670	11,745	12.15	10.78	11.46
Peat									
Kiln-dried	7.68	4,470	3,140	2,420	1,660	9,660	10.00	8.92	9.42
Air-dried, 25 per cent water	5.76	4,000	2,820	2,240	1,556	7,000	7.25	6.41	6.78
Wood—									
Kiln-dried	6.00	4,080	2,910	2,260	1,530	7,245	7.50	6.64	7.00
Air-dried, 20 per cent water	4.80	3,700	2,670	2,100	1,490	5,600	5.80	4.08	4.39

The above table gives the air required for complete combustion, the temperature attained with different proportions of air, the theoretical value, and the highest practically attainable value under a steam-boiler, assuming that the gases pass off at 320°, the temperature of steam at 75 lbs. pressure, and the incoming air at 60°; also, that with chimney draught twice, and with forced blast only, the theoretical amount of air is required for combustion.

With hickory at $5 a cord, other woods are worth about as below:

Hickory	$5 00
White oak	4 05
White ash	3 85
Apple	3 50
Red oak	4 45
White beech	3 25
Black walnut	3 25
Black birch	3 15
Hard maple	3 00
White elm	2 90
Red cedar	2 08
Wild cherry	2 75
Soft maple	2 70
Yellow pine	2 70
Chestnut	2 60
Butternut	2 55
White birch	2 40
White pine	2 10

Mr. D. K. Clark thus assigns the several portions of the heat of combustion of good coke, as burned in the locomotive:*

Making steam	10,920 B. T. U.	73 per cent.
Loss at smoke-stack	2,316 "	16.5 "
Ash and waste	764 "	5.5 "
	14,000 B. T. U.	100 per cent.

and concludes that combustion in the furnace of the locomotive may be, and often is, practically perfect, and anticipates that economy in the formation of steam will only be improved by utilizing heat now wasted at the chimney. The usual maximum evaporation is about 8 times the weight of coke used—a low figure, which is mainly due to the comparatively small proportion of heating-surface adopted. The nearer the composition of the fuel approaches that of coke, the better, as a rule, the economical effect. Coal gives, as an average, about two thirds the effect of coke, as customarily burned: and its value may be fairly approximated, the composition being known, by assuming the carbon to be the only useful constituent.

* Railway Machinery, p. 122.

Economy in combustion of fuels, where they are used simply in the production of high temperature, is so important a matter, except in those favored localities where the proximity of coal, or of peat-beds, or of forests, renders its waste less objectionable, that the engineer should omit no precaution in the endeavor to secure their perfect utilization.

To secure the greatest economy, it is necessary to adopt a form of grate which, while allowing a sufficient supply of air to pass through it to insure complete combustion, has such narrow air-spaces as to prevent waste of small fragments, by falling through them.

The narrower the grate-bars and the air-spaces, the more readily can losses from this cause and from obstruction of draught be avoided. With a hot fire, however, the difficulties arising from the warping of the bars become so great, that it is only by peculiar devices for interlocking and bracing them that their thickness can be reduced below about $\frac{3}{8}$ of an inch at the top. Many such devices are now in use. In furnaces burning wet fuel, with an ash-pit fire, fire-brick grate-bars are used.

A certain amount of air must usually be allowed to enter the furnace above the grate, to consume those combustible gases which do not obtain the requisite supply of oxygen from below. The carbon, probably, in such cases usually obtains its oxygen from below the grate, while the gaseous constituents of the fuel are consumed by the oxygen coming in above.

Chas. Wye Williams, who made most extended and careful experiments on combustion of fuel, recommended, for ordinary cases, where bituminous coal was burned, a cross area of passage, admitting air above the grate, of one square inch for each 900 pounds of coal burned per hour, or about one square centimetre for each 63 kilogrammes of fuel. This area should be made larger, proportionally, as the thickness of the bed of the fuel is increased, and as the proportion of hydrocarbons becomes greater.

Chilling the gases, before combustion is complete, should be carefully prevented; and comparatively cold surfaces, as

those of a steam-boiler, should not be placed too near the burning fuel. A large combustion-chamber should, where possible, be provided, and more complete combustion may be expected in furnaces of large size, lined with fire-brick, and with arches of the same material, than in a furnace of small size where the fire is surrounded by chilling surfaces, as in a "fire-box steam-boiler."

Finally, the greatest possible amount of heat being developed in combustion, careful provision should be made for completely utilizing that heat.

In a steam-boiler this is accomplished by having large heating-surfaces, and by so arranging the distribution of the adjacent currents of water and of hot gases that their difference of temperature shall be the greatest possible. The gases should enter the flues at that part of the boiler where the temperature is highest, and leave them at the point of lowest temperature. The feed-water should enter as near as possible to the point where the gases pass off to the chimney, and should gradually circulate until evaporation is completed at, as nearly as possible, that part of the boiler nearest to the point of entrance of the heated gases.

Where a small combustion-chamber is unavoidably employed, as in locomotives, various expedients have been devised with the object of producing complete intermixture of gases before entering the tubes. The most common and most successful is a bridge-wall, sometimes depending from the crown sheet, but sometimes rising from the grate, and which, by the production of eddies in the passing current, causes a more thorough commingling of the combustible gases with the accompanying air. None of these devices seem yet to have given such good results as to induce their general adoption.

In the furnaces of steam-boilers it is usually considered advisable to allow the gaseous products of combustion to enter the chimney at a temperature of about 600° Fahr. (315° Cent.).

The management of fires is an important but often neglected branch of instruction in fitting firemen for their special duties. The economy of boiler management is very largely dependent upon the skilful handling of the fuel and the

furnace. In general, the fires should be kept of even thickness, clear of ash and clinkers, and as clean at the sides and in the corners as elsewhere. The depth of the fuel is determined by its nature and size and by the intensity of the draught. Hard coals can be used in greater depth than soft, and large coal in deeper fuel-beds than small. A strong draught demands a thick fire, a mild draught a thin one. With a low chimney and natural draught small anthracite or fine bituminous coal may be most successfully burned in a layer but a hand's breadth in thickness; while with large "steamboat" coal of the hardest varieties and with a heavy forced draught, fires have been actually worked successfully of five times that depth, or more. The secret of success in handling fires is to find the best depth of fire for the conditions existing; to keep that thickness at all times, allowing for the ash that may accumulate; to throw the fuel on the grate at such frequent intervals as will prevent the fire burning into holes or in irregular thickness at different points; to introduce the coal so quickly and with such exactness of direction that no serious loss may occur from the inrush of cold air, and so that every shovelful should go precisely where needed, the place for the next shovelful being at the same instant located. The removal of ash is best done by means of a rake or other tool used under the grate, rather than by stirring and breaking up the bed of fuel by working through the furnace-door. The various forms of shaking grate now in use are often very efficient. For best working, the fire should usually be kept bright beneath, and the ash-pit clear. With light draught, however, and thin fires, it is sometimes advisable, if sufficient steam can be so made, to allow the fire to be less frequently raked out, and some accumulation of ash may be thus produced when working with maximum economy.

"Firing," or "stoking," as the replenishing of the fuel is called, must be done very quickly and skilfully to avoid serious annoyance by variation of steam-pressure and supply. Where several furnaces are in use this difficulty is less likely to be met with, as the fires may be cooled and cleaned in rotation. A skilful man will find it possible to keep steam very steadily with but two furnaces, even.

Ash-pits should not be allowed to become filled with ashes, as the result would be the checking of the draught, the reduction of the steaming capacity of the boiler, and loss of efficiency, even if not the melting down of the grates. It is customary at sea to clean out the ash-pits and send up ashes, throwing them overboard once in every watch of four hours, when in full steaming. If much unburned fuel is found in the ashes, it should be, if possible, cleaned out and returned to the fire, or used elsewhere.

Cleaning fires consists in thoroughly breaking up the mass of fuel on the grate, shaking out all the ashes, quickly raking out all "clinker," as the semi-fused masses of ash and fuel are called, and, after getting a level, clean bed of good fuel, as promptly as possible covering the whole with a layer of fresh coal. This is done, usually, once in four hours at sea and twice a day on land: but different fuels require somewhat different treatment. The work should be performed with the greatest possible thoroughness and dispatch, to avoid serious loss of steam-pressure.

Mr. C. W. Williams' instructions for handling the fires, where bituminous coal is used and an air-supply above the fuel is provided, are substantially as follows:

Charge the furnace from the bridge-end, gradually adding fuel until the dead-plate is reached and the whole grate evenly covered. Never permit the fire to get lower than four or five inches in thickness, of clear and incandescent fuel, uniformly distributed, and laid with especial care along the sides and in the corners. Any tendency to burn into holes must be checked by filling the hollows and securing a level surface. All lumps should be broken until not larger than a man's fist. Clean out the ash-pit so often that there shall be no danger of overheating the grate-bars.

An ash-pit, brightly and uniformly lighted by the fire above, indicates that it is in good order and working well. A dark or irregularly lighted ash-pit is indicative of an uncleaned and badly working fire. The cleaning of the fire is best done, in ordinary working, by a "rake" or other tool working on the

under side of the grates, and not by a "slice-bar" driven into the mass of fuel and above the grate.

Different fuels require different treatment. The principles just stated apply generally, but more, perhaps, to anthracite coals. The soft coals are commonly so disposed on the fire that a charge may have time to coke and its gases to burn before it is spread over the grate; liquid fuels must be so supplied that they may burn completely, at a perfectly uniform rate, and especially in such manner as to be safe from explosive combustion; the same precaution is demanded with the gaseous fuels. Special arrangements of grate and a special routine in working may be, and often are, demanded in such cases.*

The liquid and gaseous fuels are often and successfully burned in conjunction with solid fuels. In such cases the same methods are to be adopted and precautions observed in handling the latter as when burned alone.

The liquid fuels are almost invariably the crude petroleums. They are sometimes burned in a furnace in which they are allowed to drip from shelf to shelf in a series arranged vertically at the front of the furnace, the flame passing to the rear, with the entering current of air supporting their combustion. In many cases they are sprayed into the furnace by a jet of steam which should be superheated and at high pressure. The use of the steam is considered to have a peculiar and beneficial effect, possibly through chemical reactions facilitating the formation of hydrocarbons. The petroleums are all liable to cause accident if carelessly handled, and special precaution must be observed in their application to the production of steam.

The gaseous fuels are seldom used under steam-boilers, except where "natural" gas from gas-wells is obtainable, or where a very large demand or the use of metallurgical processes justifies the construction of gas-generators. Even greater precautions against accidents by explosion are needed than with the liquid fuels. In burning gas, maximum economy is secured by careful apportionment of the air-supply to the gas-consumption, and especially in avoiding excess. The regenerator system is not generally economically applicable to boilers.

The stored energy in steam and in water at any pressure

and temperature is easily ascertained by calculation, in accordance with the first law of thermodynamics.

The first attempt to calculate the amount of energy latent in the water contained in steam-boilers, and capable of greater or less utilization in expansion by explosion, was made by Mr. George Biddle Airy,* the Astronomer Royal of Great Britain, in the year 1863, and by the late Professor Rankine † at about the same time.

Approximate empirical expressions are given by the latter for the calculation of the energy and of the ultimate volumes assumed by unit weight of water during expansion, as follows, in British and in metric measures:

$$U = \frac{772(T - 212)^2}{T + 1134.4}; \qquad U_m = \frac{423.55(T - 100^2)}{T + 648};$$

$$V = \frac{36.76(T - 212)}{T + 1134.4}; \qquad V_m = \frac{2.29(T - 100)}{T + 648}.$$

These formulas give the energy in foot-pounds and kilogrammetres, and the volumes in cubic feet and cubic metres. They may be used for temperatures not found in the tables to be given, but, in view of the completeness of the latter, it will probably be seldom necessary for the engineer to resort to them.

The quantity of work and of energy which may be liberated by the explosion, or utilized by the expansion, of a mass of mingled steam and water has been shown by Rankine and by Clausius, who determined this quantity almost simultaneously, to be easily expressed in terms of the two temperatures between which the expansion takes place.

When a mass of steam, originally dry, but saturated, so expands from an initial absolute temperature, T_1, to a final absolute temperature, T_2, if J is the mechanical equivalent of the

* Numerical Expression of the Destructive Energy in the Explosions of Steam Boilers.

† On the Expansive Energy of Heated Water.

unit of heat, and H is the measure, in the same units, of the latent heat per unit of weight of steam, the total quantity of energy exerted against the piston of a non-condensing engine, by unity of weight of the expanding mass, is, as a maximum,

$$U = JT_2\left(\frac{T_1}{T_2} - 1 - \text{hyp log}\frac{T_1}{T_2}\right) + \frac{T_1 - T_2}{T_1}H.$$

This equation was published by Rankine a generation ago.*

When a mingled mass of steam and water similarly expands, if M represents the weight of the total mass and m is the weight of steam alone, the work done by such expansion will be measured by the expression

$$U = MJT_2\left(\frac{T_1}{T_2} - 1 - \text{hyp log}\frac{T_1}{T_2}\right) + m\frac{T_1 - T_2}{T_1}H.$$

This equation was published by Clausius in substantially this form.†

It is evident that the latent heat of the quantity m, which is represented by mH, becomes zero when the mass consists solely of water, and that the first term of the second member of the equation measures the amount of energy of heated water which may be set free, or converted into mechanical energy by explosion. The available energy of heated water, when explosion occurs, is thus easily measurable.

36. The Specific Heats of Water and Steam vary somewhat with temperature; this variation is noted with all solids, and occurs with the vapors, although in vastly less degree; and this is one point in which they are distinguished from the gases. For all the purposes of the engineer the specific heat of either saturated steam or of steam-gas may be taken at the value obtained by Regnault, 0.305, the quantity of heat, in thermal units, demanded to raise the temperature of units of weight of saturated steam one degree, while still keeping it saturated by

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* Steam-engine and Prime Movers, p. 387.

† Mechanical Theory of Heat, Browne's translation, p. 283.

the evaporation of additional water; which latter process demands the transformation of 0.695 unit of heat.

The specific heat of isolated steam-gas, or superheated steam, is given by Regnault as 0.48051, and constant.

The specific heat of water was determined by Regnault * very carefully and exactly, and the figures so obtained have been found capable of being very accurately represented by the following empirical formula of Rankine: †

$$C = 4 + 0.000000309(t - 39^\circ.1)^2,$$

in which t is the temperature on the common Fahrenheit scale.

The total heat demanded from t_1 to t_2 would thus be

$$h = \int_{t_1}^{t_2} C dt = t_2 - t_1 + 0.000000103[(t_2 - 39^\circ.1)^3 - (t_1 - 39^\circ.1)^3];$$

and the mean specific heat for such a range of temperature is

$$\frac{h}{t_2 - t_1} = 1 + 0.000000103[(t_2 = 39^\circ.1)^2 + (t_2 - 39^\circ.1)(t_1 - 39^\circ.1) + (t_1 - 39^\circ.1)^2].$$

On the Centigrade scale these expressions become

$$C = 1 + 0.000001(t - 4^\circ)^2,$$

$$h = t_2 - t_1 + 0.00000033[(t_2 - 4^\circ)^3 - (t_1 - 4^\circ)^3],$$

$$\frac{h}{t_2 - t_1} = 1 + 00000033[(t_2 - 4^\circ)^2 + (t_2 - 4^\circ)(t_1 - 4^\circ) + (t_1 - 4^\circ)^2].$$

The specific heat of *ice* is given by M. Person as 0.504.

Regnault's and Wiedemann's experiments, made on simple gases, and on carbonic oxide, which is formed without condensation, proved that in these cases the specific heat between 0° and 200° C. is constant; whilst their experiments on gases

* Mem. of the Academy of Sciences, 1847.

† Trans. Royal Soc. of Edinburgh, 1851. Steam-engine, p. 246.

formed with condensation show that the specific heat varies, the mean being given in the following empirical formulæ:

For CO_2	$= 44$ gr. C.	$= 8.41 + 0.0053t$	Mean of Regnault
" NO	$= 44$ "	$= 8.96 + 0.0028t$	and Wiedemann.
" C_2S_4	$= 76$ "	$= 10.62 + 0.007t$,	Regnault.
" NH_3	$= 17$ "	$= 8.51 + 0.00265t$,	Wiedemann.
" C_4H_4	$= 28$ "	$= 9.42 + 0.0115t$,	Wiedemann.

37. **The Computation of Latent and Total Heat of Steam** is readily made by means of formulas given by Regnault or based upon his work, which covered a wide range of temperature—from a little below the freezing-point to about 375° F. (190° C.). The following is the formula of Regnault for latent heat as slightly modified and corrected by Rankine for the British and metric systems, respectively:

$$l = 1091.7 - 0.695(t - 32°) - 0.000000103(t - 39°.1)^2;$$

$$l_m = 606.5 - 0.695t - 0.000000333(t - 4°)^2;$$

or, approximately, as given by the investigator,

$$l = 1092 - 0.7(t - 32°);$$
$$= 968 - 0.7(t - 212);$$
$$= 1147 - 0.7t;$$
$$l_m = 606 - 0.7t.$$

The *total heat of evaporation* is the sum of the latent and sensible heats, and may be taken as

$$h = C(t_1 - t_2) + l_1;$$
$$= 1091.7 + 0.305(t - 32°);$$
$$h_m = 606.5 + 0.305t;$$

in which the "total heat" measured is that *from* t_2 *at* t_1, the original temperature of the water and that of evaporation, and the formulas given being based on the assumption that t_1 is

taken at the melting-point of ice. For any other temperature the following will give satisfactorily exact measures:

$$h = 1092 + 0.3(t_1 - 32) - (t_2 - 32^\circ);$$
$$= 1146 + 0.3(t_1 - 212) - (t_2 - 32^\circ);$$
$$h_m = 606.5 + 0.3t_1 - t_2;$$

h being obtained in British measures and h_m in metric.

For *steam-gas*,

$$h = 1092 + 0.48(t_2 - t_1).$$

Professor Unwin proposes the following for the latent heat of vaporization of steam:

$$l_m = 799 - \frac{894}{(7.5030 - \log p)^{0.8}},$$

which is found to be extremely exact. He also obtains for the expansion during change of state,

$$\Delta v_m = 10.821 \frac{l_m}{p(7.5030 - \log p)};$$

p being given in millimetres of mercury.

38. Factors of Evaporation measure the relative amount of heat demanded to effect the heating of water from a given temperature, t_1, and its vaporization at a higher temperature, t_2, and to simply produce vaporization at the boiling-point under atmospheric pressure, which latter is now usually taken as a standard. The value of this factor of evaporation is evidently

$$f = 1 + \frac{0.3(t_1 - 212^\circ) + (212^\circ - t_2)}{966.1}, \text{ nearly.}$$

In Table XII, Appendix, are values of such factors, calculated as above by Mr. Kent.

It is seen that the relative cost of using feed-water at any one temperature as compared with the use of water at any other temperature is as the reciprocal of their factors of evapo-

rization. Thus, if feed-water can be supplied, by means of a heater, at 210° F., where previously drawn from the mains at 50°, the relative cost of making steam will be, at 100 pounds pressure, by gauge, $\frac{1044}{1208} = 0.86$, and a gain of fourteen per cent will be effected. As will be seen later, these tables are very useful in reducing the data obtained in trials of steam-boilers to the standard.

39. **Regnault's Tables** have been reproduced in many forms, usually with additions. The Appendix, among other tables, contains the data obtained by Regnault, and these values are accepted as standard universally. The table there given exhibits the temperatures and corresponding pressures of saturated steam throughout the full range now used in steam-boilers and far beyond; the quantity of heat, sensible and latent, in unity of weight; the total heat of evaporation and the density of the steam. Reference to these tables is vastly more convenient than calculation. Should it be necessary, or desirable, however, to make such calculations, the formulas already given will furnish the means. They also permit the calculation of data beyond the limits of Regnault's experiments, and are probably practically correct far beyond any pressure likely to become familiar in the operation of steam-boilers. Regnault's limit was at 230° C. (446° F.). Rankine's formula has been used beyond it.

The formulas used in these calculations are also given in the Appendix, grouped for convenience of reference. The Appendix also contains all the numerical constants ordinarily needed in computation of the quantities demanded in determining the efficiencies and performance of engines or boilers or both.

The larger works on the steam-engine and on thermodynamics include tables for use by the engineer. "Porter on the Indicator" contains a carefully revised set. Recently such tables have been issued separately. Prof. Peabody's are the latest to date. Their author adopted the latest results of Rowland in the determination of the mechanical equivalent of heat, taking its measure as 778 in British or 426.9 in metric units.*

* Tables of Properties of Saturated Steam, etc., by C. H. Peabody. New York: J. Wiley & Sons, 1888.

CHAPTER III.

RESULTS OF STANDARD METHODS; APPARATUS.

40. The Results of Trials actually conducted under acceptable conditions, and with all the precautions which have been suggested, are illustrated by the following examples:

The first case is a trial which was carried out in accordance with the above programme. The measurements of the feed-water were made by passing the water through a Worthington metre into two wooden tanks located on Fairbanks' Standard Platform Scales. The pipe connections were so arranged that one tank could be filled and weighed while the other tank was being emptied into the boiler.

Each tank was filled once every half hour. As soon as the tank was full and the pumping into the boiler commenced, the temperature of the feed-water was taken by sensitive thermometers reading to one-tenth of a degree.

All water measurements, as in all instances of careful work, were made by weight rather than by volume, and systems of checking were devised and practised whenever practicable. The apparatus was all carefully standardized and repeatedly re-examined and tested as opportunity offered.

The measurements of the coal were effected by weighing the coal previous to its being wheeled into a pile in the coal-room. The second weighing was made when the coal was fed into the furnace. As far as it was possible, the furnace was supplied with coal at intervals of every half hour, so as to correspond as nearly as could be to the feeding of the water.

After the completion of the test, a careful analysis of the coal was made, to determine upon a sufficiently large scale its calorific power and the quantity of contained moisture. The

steam from the boiler was condensed by means of a continuously acting calorimeter, formed by placing four tanks on Fairbanks Standard Platform Scales. The steam from the boiler was passed through a surface-condenser having a condensing surface of 631 sq. ft. As fast as the steam was condensed from the boiler it was received in small tanks located on platform-scales. These tanks were similar in size to the feed-water tanks, and were so arranged as to be filled and emptied once every half hour, one tank receiving the condensed water from the boiler while the other was being emptied.

The condenser was supplied with a large volume of cold water from a weir just outside of the works, and after flowing through the condenser and thereby cooling the steam and receiving therefrom the contained heat, this water was caught in two large tanks placed on platform-scales. These tanks were also arranged so that one tank could be emptied while the other was being filled, and were of sufficient capacity so as to insure catching all of the water required for half an hour's run in the condenser. The temperature of the inlet water of the condenser, of the outlet water, and of the condensed steam were carefully noted by means of thermometers reading to a tenth of a degree. Readings of the inlet water and of the condensed steam were taken once every half hour at the same time that the quantities of the water in the tanks were weighed. Inasmuch as the outlet to the condenser varied considerably in temperature, readings on this were taken every five minutes during the entire time of the test. It will thus be seen that a very correct average of the amount of heat given to the condenser was obtained. The quantity of air supplied by the blowers to the furnace was measured by continuously acting anemometers placed in the supply-pipes. The readings of the anemometers were checked by means of the number of revolutions of the blowers and their cubic feet per revolution.

The steam-pressure was kept by a recording pressure-gauge, which was checked by an exceedingly delicate and sensitive gauge, which previously, and subsequently to the test,

was carefully verified by means of a mercury column. Constant records of the hygrometer, barometer, and thermometers, both in the boiler-room and of the external air, were kept during the entire period of the test.

It will be seen from the above, that all of the processes and measurements were kept in duplicate in such a way as to afford a constant check on each other and preclude the possibility of any errors.

Samples of steam were taken in a small calorimeter for the purpose of ascertaining whether the boiler supplied wet steam.

The following is a brief condensed summary:

EFFICIENCY AS PER TEST, 7.50 A.M. to 7.50 A.M.

Total heat of boiler	64,536,613	heat-units.	
Steam	42,933,141	" "	66.6 per cent.
Heat escaping in flue-gases	9,669,036	" "	15 " "
Radiated heat	5,162,939	" "	8 " "
Heat to vaporize moisture in coal	141,372	" "	0.2 " "
Heat to vaporize moisture in air supplied to furnace	345,978	" "	0.4 " "
Leakage	3,531,645	" "	4.0 " "
" from pump	127,936	" "	0.2 " "
Heat absorbed by fire-brick	2,581,645	" "	4.0 " "
Unaccounted for	1,092,941	" "	1.6 " "

In the trial of an upright boiler reported on by Sir Frederick Bramwell, in 1876, coke being used as the fuel and wood in starting the fires, the following data* were obtained:

Ash and moisture	43.79 lbs.
Combustible	194.46 "
Total fuel	238.25 "
Air used per pound combustible	17½ "
Heat generated, net	2,798,312 B. T. U.
" " per lb. fuel	11,745 " "
" " available, net	2,101,700 " "
Water evaporated	1,620 lbs.
The efficiency of the furnace was	0.643 "

* Conversion of Heat into Work. Anderson.

The balance-sheet stands thus:

Dr.

	Available heat	2,101,700	B. T. U.
	Cr.		
Per Cent.			
88.29	Heat expended in evaporation	1,855,900	B. T. U.
7.03	Displacing atmosphere	147,720	" "
3.35	Loss by conduction and radiation	70,430	" "
.05	Heat in ashes	1,129	" "
1.26	Unaccounted for	25,521	" "
100.00		2,101,700	

In trials conducted by the Author, for a committee of the American Institute, of which he was chairman, in testing a number of different types of boiler,* a surface-condenser was employed to condense *all* steam made, and results thus for the first time obtained which gave exact measures of net efficiency, the quality of all steam made being determined.

In calculating the results from the record of the logs, the committee first determined the amount of heat carried away by the condensing water by deducting the temperature at which it entered from that at which it passed off. To this quantity is added the heat which was carried away by evaporation from the surface of the tank, as determined by placing a cup of water in the tank at the top of the condenser at such height that the level of the water inside and outside the cup were the same, noting the difference of temperatures of the water in the cup and at the overflow, and the loss by evaporation from the cup. The amount of evaporation from the surface of the water in the cup and in the condenser, which latter was exposed to the air, was considered as approximately proportional to the tension of vapor due their temperatures, and was so taken in the estimate. The excess of heat in the water of condensation over that in the feed-water also evidently came from the fuel, and this quantity was also added to those already mentioned.

* See Transactions, 1871; also, Report on Mechanical Engineering at Vienna International Exhibition, 1873, R. H. T.

The total quantities were, in thermal units, as follows:

A	34,072,058.09
B	48,241,833.60
C	24,004,601.14
D	38,737,217.57
E	11,951,002.10

These quantities, being divided by the weight of combustible used in each boiler during the test, will give a measure of their relative economical efficiency; and, divided by the number of square feet of heating-surface, will indicate their relative capacity for making steam. But as it was the intention of the committee to endeavor to establish a practically correct measure that should serve as a standard of comparison in subsequent trials, it was advisable to correct these amounts by ascertaining how and where errors have entered, and introducing the proper correction. There were two sources of error that are considered to have affected the result as above obtained. The tank being of wood, a considerable quantity of water entered it, leaked out again at the bottom, without increase of temperature, instead of passing through the tank and carrying away the heat, as it is assumed to have done in the above calculation. The meters also registered rather more water than actually passed through them, and this excess assists in making the above figures too high. The sum of these errors the committee estimated at 4 per cent of the total quantity of heat carried away by the condensing water. The other two quantities were considered very nearly correct.

Making these deductions, we have the following as the total heat, in British thermal units, which was thrown into the condenser by each boiler:

A	32,751,835.34
B	46,387,827.10
C	23,066,685.39
D	37,228,739.07
E	11,485,777.35

That the figures thus obtained are very accurate is shown by calculating the heat transferred to the condenser by the steam-boilers denominated A and B both of which superheated their

steam), by basing the calculation on the temperature of the steam in the boiler, as given by the thermometer, the results thus obtained being 32,723,681.76 and 46,483,322.5, respectively.

Dividing these totals by the pounds of combustible consumed by each boiler, we get as the quantity of heat per pound, and as a measure of the relative economic efficiency:

A	10,281.53
B	10.246.92
C	10,143.66
D	10,048.24
E	10,964.94

Determining the weight, in pounds, of water evaporated per square foot of heating-surface per hour, we get as a measure of the steaming capacity:

A	2.65
B	3.59
C	2.83
D	3.10
E	1.92

The quantity of heat per pound of combustible, as above determined, being divided by the latent heat of steam at 212° Fahrenheit (966°.6), gives as the equivalent evaporation of water at the pressure of the atmosphere, and with the feed at a temperature of 212° Fahrenheit:

A	10.64
B	10.60
C	10.49
D	10.40
E	10.34

For general purposes this is the most useful method of comparison for economy.

The above figures afford a means of comparison of the boilers, irrespective of the condition (wet or dry) of the steam furnished by them. All other things being equal, however, the committee consider that boiler to excel which furnishes the driest steam; provided that the superheating, if any, does not exceed about 100°.

In this trial the superheating was as follows:

A	16°.08
B	13°.23
C	0.
D	0.
E	0.

As the boilers C, D, E did not superheat, it became an interesting and important problem to determine the quantity of water carried over by each with the steam. This we are able, by the method adopted, to determine with great facility and accuracy.

Each pound of saturated steam transferred to the condensing water the quantity of heat which had been required to raise it from the temperature of the water of condensation to that due to the pressure at which it left the boiler, *plus* the heat required to evaporate it at that temperature. Each pound of water gives up only the quantity of heat required to raise it from the temperature of the water of condensation to that of the steam with which it is mingled. The total amount of heat is made up of two quantities, therefore, and a very simple algebraic equation may be constructed which shall express the conditions of the problem:

Let

H = heat-units transferred per pound of steam.
h = heat-units transferred per pound of water.
U = total quantity of heat transferred to condenser.
W = total weight of steam and water, or of feed-water.
x = total weight of steam.
$W-x$ = total weight of water primed.

Then

$$Hx + h(W - x) = U; \text{ or } x = \frac{\frac{U}{h} - W}{\frac{H}{h} - 1}.$$

Substituting the proper values in this equation, we deter-

mine the absolute weights and percentages of steam and water delivered by the several boilers as follows:

	Weight of Steam.	Weight of Water.	Percentage of Water Primed to Water Evaporated.
A	27,896.	0.	0.
B	39,670.	0.	0.
C	19.782.94	645.06	3.26
D	31.663.35	2,336.65	6.9
E	9,855.6	296.9	3.

And the amount of water, in pounds, actually evaporated per pound of combustible:

A	8.76
B	8.76
C	8.70
D	8.55
E	9.41

Comparing the above results, the committee were enabled to state the following order of capacity and of economy in the boilers exhibited, and their relative percentage of useful effect, as compared with the economical value of a steam-boiler that should utilize all of the heat contained in the fuel:

	Steaming Capacity.	Economy of Fuel.	Percentage of Economical Effect.
A	No. 4	No. 2	0.709
B	No. 1	No. 3	0.707
C	No. 3	No. 4	0.699
D	No. 2	No. 5	0.693
E	No. 5	No. 1	0.756

The results obtained as above, and other very useful determinations derived from this extremely interesting trial, were given in the table, as a valuable standard set of data with which to compare the results of future trials, and as a useful aid in judging of the accuracy of statements made by boiler-venders in the endeavor to effect sales by presenting extravagant claims of economy in fuel.

Mr. Druitt Halpin found the following net results of test of a variety of English-built boilers:

No.	Description of Boiler.	Pounds Water Evaporated. Per square foot of heating-surface per hour	Per pound of fuel from and at 212 degrees.	Thermal Units. To fuel.	Transmitted per hour per sq. ft. heating-surface per hour	Per pound fuel.	Efficiency.	Figure of merit = units per sq. ft. per hour × efficiency
1	Field	4 57	8 83	....	4,414	8.529	.	
2	Field	2 28	10 83	.	2,203	10,461	.	
3	Field	2 57	10 93	..	2,482	10,558		
4	Portable } Cardiff	1 52	10 23	14,718	1,468	9,882	67	98,356
5	Portable } Cardiff	2 26	10 49	14,718	2,183	10,133	68	148,444
6	Portable } Cardiff	1 76	11 81	14,718	1,700	11 408	77	130,900
7	Portable } Cardiff	3 56	9 93		3,438	9,592		118 248
8	Lancashire	1 57	12 83	15,715	1,516	12,393	77	108.248
9	Lancashire	2 83	9 89	13 833	2,733	9 553	68	185,844
10	Lancashire	1 88	12 25	15,715	1 816	11,833	75	136 200
11	Jacketed	4 70	7 7	14 805	4,595	7 500	50	229 750
12	Lancashire	2 57	10 9	15,715	2,482	10.579	67	166 294
13	Compound	1 43	11 51	14,296	1,381	11,125	78	107 718
14	Loco. (Webb)	9 83	10 28	14.004	9,495	9,930	70	664 650
15	Loco (Marie)	4 62	10 65	14,600	4,462	10,287	70	312,340
16	Loco. } Coke	12 57	8 22	13,550	12,142	7,940	58	704,236
17	Loco } Coke	13 73	8 94	13,550	13,263	8,[illegible]36	63	835 569
18	Loco } Coke	6 76	10 01	13,550	6,530	9,609	71	463,630
19	Loco } Coke	7 39	11 2	13.550	7,138	10.819	77	549,626
20	Torpedo	12 54	8 37	14,727	12,113	8 085	54	654,102
21	Torpedo	14 86	7 78	14,727	14,354	7,523	51	732,054
22	Torpedo	17 90	7 49	14,727	17,291	7,235	49	847 259
23	Torpedo	20 74	7 04	14,727	20,034	6,800	46	921,564
24		a	b	c	d	e	f	g

The "locomotive" boiler is found to be more efficient as a part of the engine and on the track than when mounted as a stationary boiler, an unexpected result.

41. The Quality of Steam made in any boiler, or as supplied to an engine, is hardly less important than the quantity. When the steam is required for heating purposes simply, or even when all the heat issuing as waste, necessary or other, from the exhaust-ports of a non-condensing engine cylinder can be utilized for useful and paying purposes, this is a matter of no importance; but when it is essential that loss in the engine shall be made a minimum, and that the engine shall have maximum efficiency, the quality of the steam becomes exceedingly important. Dry steam is very much more efficient as a working substance in the steam-engine than wet; since, where the latter is supplied from the boiler, the waste by cylinder-condensation is greatly increased -and so greatly that the more obvious direct loss by the passing of heat through the engine in unavailable form, hot water acting as its vehicle, becomes comparatively small. The determination of the quality

of steam by any boiler is thus as important as the measure of its apparent evaporation.

The difference between the apparent and the actual evaporation is often very great. A good boiler properly managed will usually "prime" less than five per cent, even though having no superheating-surface, and less than two per cent may usually be hoped for. Steam is often made practically dry. But a hard-worked boiler, or one having defective circulation, will often prime ten or twenty per cent; and cases have been found in the experience of the Author in which the quantity of water carried out of the boiler by the current of steam exceeded the weight of the steam itself. It has thus happened that, where no measure of this defect has been made, the apparent evaporation only being reported, the quantity of water said to have been evaporated has equalled, and sometimes has even greatly exceeded, the theoretically possible evaporation of an absolutely perfect boiler. It is thus essential that, when the apparent evaporation has been determined by trial, the quantity of water entrained with the steam be measured and deducted, and then *real* evaporation thus ascertained and reduced for the standard conditions. Under ordinarily good conditions, a real evaporation of ten or eleven times the weight of the fuel, corresponding to an efficiency of 0.75 to 0.80, represents the best practice, and a real evaporation of twelve of water by one of combustible, from and at the boiling-point, or an efficiency of eighty per cent, is rarely observed under the usually best conditions of steam-boiler practice. Where more than the efficiency here given as probable is reported, the work should be very carefully revised, and errors sought until absolute certainty is secured.

Trials not including calorimetric measurement of the water entrained with the steam are comparatively valueless, and should be rejected in any important case. Reports of extraordinary economy are often based on this kind of error. The experiments of M. Hirn at Mulhouse showed an average of about 5 per cent priming; Zeuner makes it approximately from 7½ to 15 per cent; while the experiments of the Author at the American Institute in 1871 give from 3 to 6.9 per cent.

A recently devised method of measuring the amount of moisture in the steam is to introduce into the boiler with the feed-water sulphate of soda, and at intervals to draw from the lower gauge-cock a small amount of water, and also from the steam, condensing either by a coil of pipe in water or a small pipe in air. A chemical analysis gives the proportion of sulphate of soda in each portion, and the quotient of the proportion of sulphate of soda in the portion from the steam by the proportion in that from the water gives the ratio of water entrained, as steam does not carry sulphate of soda, which is only brought over by the hot water entrained. This method was used by Professor Stahlschmidt at the Düsseldorf Exhibition Trials.

42. The Calorimeters used in determining the quantity of moisture in steam have several forms, widely differing in construction, and to some extent in value. They nearly all embody the same principles, however. The objects sought to be attained in their construction are: The exact measurement of the weight of steam received by them from the boiler, and of its temperature and pressure at the boiler; the determination of the weight of water used in its condensation and the range of temperature through which it is raised in the operation; the reduction of wastes of heat in the calorimeter to a minimum, and the exact measurement of that waste if it is sensibly or practically noticeable.

The Barrel or Tank Calorimeter as employed by the Author, is the simplest form of this instrument which has been produced. It consists of a strong barrel or tank, of hard wood, absorbing little of either water or heat, and having a movable cover. This tank is mounted on platform-scales capable of accurate adjustment and having as fine readings as possible. It is filled with water to within about one fourth its height from the top, and the steam is led into it through a rubber tube or hose of sufficient capacity to supply the steam to the amount of one eighth or one tenth the weight of the water in three or five minutes. A steam-gauge of known accuracy gives the boiler-pressure, and the corresponding temperature and total heat of the steam are ascertained from the steam-tables.

In using this apparatus the steam is rapidly passed into the mass of water contained in the tank, until the scales show that the desired quantity has been added. The steam is so directed by varying the position of the end of the tube, and by inserting it so deeply in the water that the whole mass is very thoroughly stirred, and a very perfect mixture secured of condensing water with the water of condensation; and so that the temperatures indicated by the inserted thermometer shall be the real mean temperature of the mass. The weights and

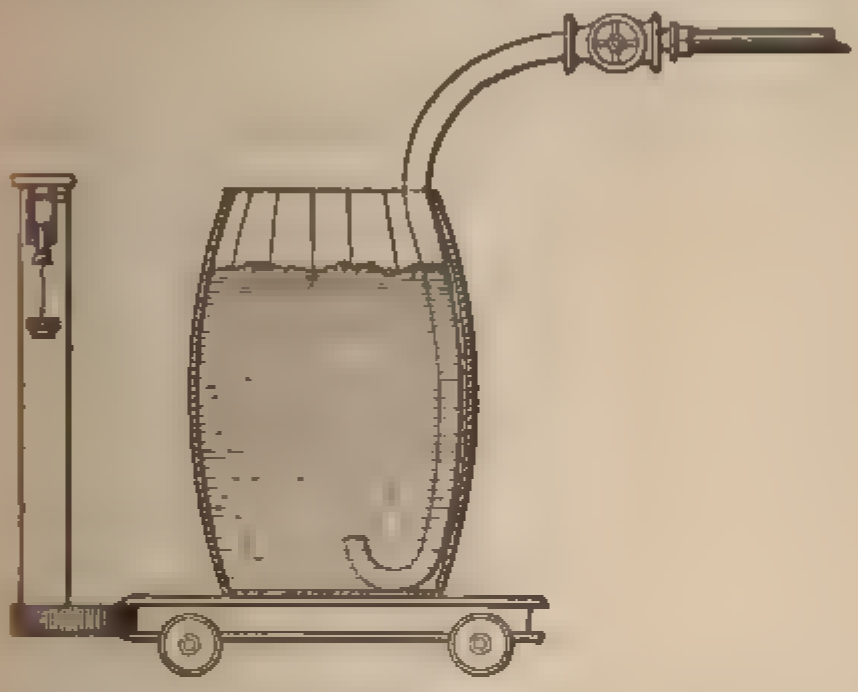

FIG. 1.—THE CALORIMETER.

temperatures are then inserted in the log of the trial, as below, and the proportion of water brought over with the steam is thence easily calculable. The thermometers employed usually read to tenths of a degree Fahrenheit, or to twentieths of a centigrade degree, accordingly as the one or the other scale is employed. Readings must be made with the greatest possible accuracy, and in sufficient number to insure a satisfactorily exact mean. With good thermometers and scales, a reliable gauge, and care in operation, good results can be obtained by averaging a series of trials.*

The *Hirn Calorimeter* is substantially the same as the above, with the addition of an apparatus for stirring the water

* Report on Boiler Trial, Trans. A. S. M. E. 1884, vol. vi.

in the tank to insure thorough mixture and readings of temperature of condensing water exactly representative of the true mean temperature of the mass after the introduction of the steam. This is not an essential feature of the apparatus, if the Author may judge by his own experience, provided the jet of entering steam is so directed as to cause rapid circulation. No stirring apparatus could operate more efficiently than the force of the steam itself, properly directed. Hirn was probably the first (1868) to attempt the determination of the quality of steam as delivered from steam-boilers.* A similar apparatus was used at the trials of the Centennial International Exhibition, Philadelphia, 1876.†

43. **The Theory of the Calorimeter** is as follows:‡

Each pound of saturated steam transferred to the condensing water the quantity of heat which had been required to raise it from the temperature of the water of condensation to that due to the pressure at which it left the boiler, *plus* the heat required to evaporate it at that temperature. Each pound of water gives up only the quantity of heat required to raise it from the temperature of the water of condensation to that of the steam, with which it is mingled. The total amount of heat is made up of two quantities, therefore, and a very simple algebraic equation may be constructed, which shall express the conditions of the problem:

Let, as in § 258,

H = heat-units transferred per pound of steam;
h = heat-units transferred per pound of water;
U = total quantity of heat transferred to condenser;
w = total weight of steam and water, or of feed-water;
W = condensing water,
x = total weight of steam;
$w - x$ = total weight of water primed.

* Bulletin de la Société Industrielle de Mulhouse, 1868–9.
† Reports of Judges, vol. vi.
‡ First published by the Author, who had not then become aware of the work done by M. Hirn, in Trans Am. Inst. Report on Boiler Trial, 1871. See also Vienna Reports, vol. iii. p. 123.

Then

$$Hx + h(w - x) = U; \text{ or } x = \frac{\frac{U}{h} - W}{\frac{H}{h} - 1} = \frac{U - wh}{H - h}.$$

Substituting the proper values in this equation, we determine the absolute weights and percentages of steam and water delivered by the boiler.

Or, let

Q = quality of the steam, dry saturated steam being unity;
H' = total heat of steam at observed pressure;
T = " " " water " "
h' = " " " condensing water, original;
h_1 = " " " " " final.

And we have the equivalent expression, as written by Mr. Kent,

$$Q = \frac{1}{H' - T}\left[\left(\frac{W}{w}(h' - h_1)\right) - \left(T - h_1\right)\right].$$

The value of the quantity U is obtained by multiplying the weight of water in the calorimeter originally by the range of temperature caused by the introduction of the steam from the boiler. Mr. Emery employs another form, as below, in which Q is the quality of steam as before; W the weight of condensing water; w the weight added from the boiler; T the temperature due the steam-pressure in the boiler; t the initial and t_1 the final temperature of the calorimeter; l the latent heat of evaporation of the boiler-steam; and x the weight of steam corresponding to l. Thus,

$$x = \frac{W(t_1 - t) - w(T - t_1)}{l}, \qquad y = 100\frac{w - x}{w};$$

and

$$Q = \frac{x}{w} = \frac{W(t_1 - t) - w(T - t_1)}{lw}.$$

The following expressions give the quality of steam as computed from metric data supplied by the calorimeter, and Bosscha's corrections being introduced for specific heat of water at varying temperatures: *

w_1 = weight of condensing water;
p_1 = absolute pressure of steam;
t_1 = the initial temperature of condensing water;
t_2 = the final temperature of the water;
t_3 = the temperature of steam in boiler;
w_2 = cold-water weight;
w_3 = weight of steam condensed;
x' = percentage of steam in the mixture from the boiler, uncorrected for specific heat of water;
x = same as x', but with Bosscha's correction;

$$x' = \frac{(t_2 - t_1)w_2 - (t_3 - t_2)w_3}{(606.5 - 0.695t_3)w_3},$$

$$x = \frac{(t_2 - t_1)w_2 - (t_3 - t_2)w_3 - 0.00011[(t_2^2 - t_1^2)w_2 - (t_3^2 - t_2^2)w_3]}{(606.5 - 0.695t_3)w_3}.$$

Mr. Nystrom has employed the Hirn calorimeter, substituting *ice* for cold water as a condensing medium.† In this case, adopting his notation,

w = pounds of cold water put into the barrel;
h = units of heat per pound of w when cold and above 32°;
I = pounds of ice put into the barrel;
W = pounds of heated water in the barrel after the completion of the experiment; that is, including the weight of the condensed steam;
h' = units of heat per pound of W above 32°;
f = pounds of foam or water carried over with the steam into the barrel;
S = pounds of saturated steam blown into the barrel;

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* Proc. Brit. Inst. C. E., 1888, No. 2306.

† Pocket-book; Humidity of Steam, p. 572.

$H =$ units of heat per pound of the steam S;
$H' =$ units of heat per pound of the foam f;
$p =$ pounds of steam and foam carried over from the boiler into the barrel;
$P =$ units of heat passed over with the steam and foam into the barrel;

The weight p must then be equal to the sum of the weights of the steam S and foam f, which is evidently the same as the difference between the weights W and w.

That is,

$$p = S + f = W - w.$$

The total units of heat P passed over with the steam S and foam f must then be:

$$P = HS + H'f = Wh' - wh.$$

By solving this formula for the steam S, we have:

$$S = \frac{P - H'f}{H} = W - w - f.$$

$$S = p - f \because$$

$$H(p - f) = P - H'f;$$

$$Hp - Hf = P - H'f;$$

$$f(H - H') = Hp - P.$$

From this formula we have the weight of foam carried over with the steam from the boiler into the barrel; namely,

$$f = \frac{Hp - P}{H - H'}.$$

But $P = Wh' - wh$, which, inserted in the formula, gives:

Pounds of foam, $$f = \frac{Hp + wh - Wh'}{H - H'}. \quad (8).$$

The percentage of humidity of the steam will then be:

$$\% = \frac{100f}{p}.$$

The Formula (8) is ready for use of the data obtained by the calorimeter when $p = W - w$, and when no ice is used.

For the melting of 1 pound of ice requires 142.65 units of heat, according to Regnault's delicate experiments, but for the caloric experiments on humidity of steam 142.6 units of steam will be more correct. Then the units of heat required to melt the ice to water of 32° by the steam in the barrel will be $142.6I$, and the heat required to raise the temperature of that water from 32° to that of W when the experiment is completed will be Ih'. But the weight of ice melted to water is included in the weight W; the heat passed with the steam from the boiler into the ice and cold water will be,

$$P = Wh' - wh + 142.6I.$$

This formula, inserted for P in Formula (7), will give the weight of foam passed with the steam from the boiler into the ice and cold water.

$$f = \frac{Hp + wh - (Wh' + 142.6I)}{H - H'}.$$

When no cold water is used, but the humid steam is blown into only ice in the barrel, then the weight of foam will be:

$$f = \frac{Hp - (Wh' + 142.6I)}{H - H'}.$$

The percentage of humidity will be in either case

$$\% = \frac{100f}{p}.$$

For humid steam, $Hp > P$.

For saturated steam, $Hp = P$.

For superheated steam, $Hp < P$.

When the steam is superheated, the formulas give a negative value of f.

The following data and results are given in illustration of this method by its author:

Example.—Steam-pressure by gauge, 98 pounds.

$$H = 1184.6 \text{ and } H' = 308.7.$$

Weight of empty barrel with top, ... 64.25 lbs.
$I = 80.5$ pounds of ice, ... 144.75 "
$w = 287.25$ pounds of water at $71°$ $h = 39.015$, 432.00 "
$W = 404$ pounds of mixture at $136°$ $h = 104.2$, 468.25 "
$p = 36.25$ pounds of steam and foam.

Formula 11.

$$f = \frac{1184.6 \times 36.25 + 287.25 \times 39.015 - (404 \times 104.2 + 142.6 \times 80.5)}{1184.6 - 308.7}$$
$$= 0.653 \text{ lbs.}$$

Humidity of the steam, $\% = \dfrac{100 \times 0.653}{36.25} = 1.8$ per cent.

In the use of the calorimeter it may be reasonably expected that errors may be made, by careful work, something less than one per cent. The results of good work should agree within less than one-half per cent. Thermometers should read within one-tenth of a degree; steam-gauges should be correct within less than one or two pounds, and weights within one per cent or less. The latter are the most trying quantities to measure with satisfactory accuracy. Generally the lower the initial temperature of the calorimeter the better the results, and it is often thought advisable to cool the condensing water, by the use of ice, down to the melting point. The amount of steam introduced should be as great as is possible without causing so high a final temperature as to cause the production of troublesome quantities of vapor in the calorimeter.

If Q exceeds unity, the steam is superheated by the amount

$$\frac{(Q-1)l}{0.48} = 2.0833l\,(Q-l);^{*}$$

and if less than unity, the priming is, in per cent, $100\,(1-Q)$.

44. Records of calorimetric tests should be even more carefully and more frequently made than in any other part of the work of a boiler-trial. The following, from work conducted by the Author, illustrates the method. The symbols relate to the first of the above formulas.

PRIMING TESTS.

Steam-pressure.	W Weight Condensing Water	w Weight Wet Steam added	T Temperature of Cold Water	T' Resultant Temperature.	Total heat in Dry Steam.	$H-T'$	$\frac{W}{w}$	$T'-T$	l	% Moisture.
100	290	10	50.8	83.4	1185.0	1001.6	29	32.6	875.4	.06
100	290	27 5	50.8	138.8	1185.0	1046.2	10 5	88	875.4	.13
100	327.5	10	55.8	85.8	1185.0	1099.2	32.75	30	875 4	.13
100	327 5	15	55.8	100.2	1185.0	1084 8	21.83	44.4	875.4	.13
100	332 0	10	99.2	125.6	1185 0	959.4	33.2	26.4	875 4	.09
100	332.0	15½	99.2	139.2	1185.0	945.8	21.1	40.0	875.4	.10
100	315.0	15	56.2	102.8	1185.0	1082.2	21.0	46.6	875 4	.11
100	315.0	25	56.2	130.0	1185.0	1055.0	12.6	73.8	875.4	.13

The boiler was a fire-tubular boiler, which was not so handled as to give as dry steam as was desired; and one object of the trial, of which the above is a part of the record, was to ascertain how seriously was the quality of the steam affected. It is seen that the priming amounted to ten or twelve per cent, with fairly uniform figures through the period of test. The steam should have entrained less than one half this proportion, had the boiler been all that was expected of it.

Errors of small magnitude, absolutely, may greatly affect the results of calculation, as is well illustrated by the following example presented by Mr. Kent:

* Centennial Report, pp. 138–9.

Assume the values of the quantities to be, as read, column 1:

	OBSERVED READING.	TRUE READING.	AMOUNT OF ERROR.
Weight of condensing water, corrected for equivalent of apparatus,* W	200.5 lbs.	200 lbs.	½ pound.
Weight of condensed steam, w	9.9 "	10.0 "	$\frac{1}{10}$ "
Pressure of steam by gauge P	78. "	80 "	2 pounds.
Original temperature of condensing water, t	44°.5 "	45° "	½ degree.
Final temperature of condensing water, t'	100°.5 "	100° "	½ "

Then let it be assumed that errors of instruments or of observation have led to the recording of slightly different figures from the true quantities, as given in column 2:

		Moisture per cent.	Error per cent.
Substituting in the formula the "true readings," we have for the value of	$Q = 0.9874$	$= 1.26$	$= 0.$
All readings true except $W = 200.5$,	$Q = .9906$	$= 0.94$	$= 0.32$
" " " " $w = 9.9$,	$Q = 1.0000$	$= 0.00$	$= 1.26$
" " " " $P = 78.0$,	$Q = .9880$	$= 1.20$	$= 0.06$
" " " " $t = 44.5$,	$Q = .9989$	$= 0.11$	$= 1.15$
" " " " $t' = 100.5$,	$Q = .9994$	0.06	$= 1.20$
" " incorrect	$Q = 1.0272$	$=$ (minus)	$= 3.98$

The last case is equivalent to 50.2 degrees superheating.

Errors of 0.1 or even 0.25 per cent in weights and of temperature of equal amount not infrequently occur, probably, where ordinary instruments are employed. The errors due to false weight in measurement of the condensed steam are liable to be very serious, and it is only by making a considerable number of observations and obtaining the mean that results can be secured, ordinarily, of real value.

45. The "Coil Calorimeter" has been devised to secure more exact results in the weighing of the water of condensation than can be obtained when it is weighed as part of the larger mass. In this instrument a coil of pipe is introduced into the tank and serves as a surface-condenser in which the boiler-steam is received and condensed, and from which it is transferred to another vessel in which it is weighed by itself with scales constructed to weigh such small weights with accuracy; or the coil is removed and weighed with the contained water. In the

* Correction made only for coil calorimeter to be described.

former case, drops of water may adhere to the internal surfaces of the coil and escape measurement; in the latter, the weight to be determined is increased by the known weight of the coil, and less delicacy of weighing becomes possible.

The following is Kent's description of his calorimeter, which is of this class, and has been found to give good results: *

A surface-condenser is made of light-weight copper tubing $\frac{3}{8}''$ in diameter and about 50′ in length, coiled into two coils, one inside of the other, the outer coil 14″ and the inner 10″ in diameter, both coils being 15″ high. The lower ends of the coils are connected by means of a brazed T-coupling to a shorter coil, about 5′ long, of 2″ copper tubing, which is placed at the bottom of the smaller coil and acts as a receiver to contain the condensed water. The larger coil is brazed to a $\frac{3}{8}''$ pipe, which passes upward alongside of the outer coil to just above the level of the top of the coil and ends in a globe-valve, and a short elbow-pipe which points outward from the coil. The upper ends of the two $\frac{3}{8}''$ coils are brazed together into a T, and connected thereby to a $\frac{3}{8}''$ vertical pipe provided with a globe-valve, immediately above which is placed a three-way cock, and above that a brass union ground steam-tight. The upper portion of the union is connected to the steam-hose, which latter is thoroughly felted down to the union. The three-way cock has a piece of pipe a few inches long attached to its middle outlet and pointing outward from the coil.

A water-barrel, large enough to receive the coil and with some space to spare, is lined with a cylindrical vessel of galvanized iron. The space between the iron and the wood of the barrel is filled with hair-felt. The iron lining is made to return over the edge of the barrel, and is nailed down to the outer edge so as to keep the felt always dry. The barrel is furnished also with a small propeller, the shaft of which runs inside of the inner coil when the latter is placed in the barrel. The barrel is hung on trunnions by a bail by which it may be raised for weighing on a steelyard supported on a tripod and lifting lever. The steelyard for weighing the barrel is graduated

* Trans. Am. Soc. M. E. 1884.

to tenths of a pound, and a smaller steelyard is used for weighing the coil, which is graduated to hundredths of a pound.

In operation, the coil, thoroughly dry inside and out, is carefully weighed on the small steelyard. It is then placed in the barrel, which is filled with cold water up to the level of the top of the globe-valves of the coil and just below the level of the three-way cock, the propeller being inserted and its handle connected. The barrel and its contents are carefully weighed on the large steelyard; the steam-hose is connected by means of its union to the coil, and the three-way cock turned so as to let the steam flow through it into the outer air, by which means the hose is thoroughly heated; but no steam is allowed to go into the coil. The water in the barrel is now rapidly stirred in reverse directions by the propeller and its temperature taken. The three-way cock is then quickly turned, so as to stop the steam escaping into the air and to turn it into the coil; the thermometer is held in the barrel, and the water stirred until the thermometer indicates from five to ten degrees less than the maximum temperature desired. The globe-valve leading to the coil is then rapidly and tightly closed, the three-way cock turned to let the steam in the hose escape into the air, and the steam entering the hose shut off. During this time the water is being stirred, and the observer carefully notes the thermometer until the maximum temperature is reached, which is recorded as the final temperature of the condensing water. The union is then disconnected and the barrel and coil weighed together on the large steelyard, the coil is then withdrawn from the barrel and hung up to dry thoroughly on the outside. When dry it is weighed on the small scales. If the temperature of the water in the barrel is raised to 110° or 120° the coil will dry to constant weight in a few minutes. After the weight is taken, both globe-valves to the coil are opened, the steam-hose connected, and all of the condensed water blown out of the coil, and steam allowed to blow through the coil freely for a few seconds at full pressure. When the coil cools it may be weighed again, and is then ready for another test.

If both steelyards were perfectly accurate, and there were no losses by leakage or evaporation, the difference between the

original and final weights of the barrel and contents should be exactly the same as the difference between the original and final weights of the coil. In practice this is rarely found to be the case, since there is a slight possible error in each weighing, which is larger in the weighing on the large steelyard. In making calculations the weights of the coil on the small steelyard should be used, the weight on the large steelyard being used merely as a check against large errors.

The late Mr. J. C. Hoadley constructed exceedingly accurate apparatus of the "coil" type and obtained excellent results.

It is evident that this calorimeter may be used continuously, if desired, instead of intermittently. In this case a continuous flow of condensing water into and out of the barrel must be established, and the temperature of inflow and outflow and of the condensed steam read at short intervals of time.

46. **The Continuous Calorimeter** is an instrument in which the operations of transfer of steam to the instrument and its examination are not intermitted, as is necessarily the case in the more commonly employed forms of the apparatus. The instrument being thus kept in use continuously, every variation in the quality of steam can be observed and the number of observations can be increased to any desired extent, and, the apparatus being accurate, any degree of exactness of mean results can be attained.

One of the earliest forms of this instrument was devised by Mr. John D. Van Buren, of the U. S. N. Engineers, and Instructor in Engineering at the Naval Academy, about 1867. This instrument, as constructed by Mr. T. Skeel, and used by a committee of judges* at the exhibition of the American Institute, 1874 5, of which the Author was chairman, was made as follows:

Steam was drawn from the steam-drum, near the safety-valve, through a felted pipe 1½ inches (3.8 cm.) diameter, into a rectangular spiral or coil consisting of 80 feet (24.4 m.) of pipe of similar size. Condensing water from the street-main was led into the tank surrounding the coil or "worm," and

* Trans. Am. Inst. 1875; Van Nostrand's Mag. 1875.

issued at the bottom through a "standard orifice," the rate of discharge from which had been determined and the law of its variation with change of head ascertained. The quantity of condensing water thus became known by observing the head of water within the tank. The water of condensation from the coil was caught in a convenient vessel, and weighed on scales provided for that purpose. The temperature of the condensing water at entrance and exit was shown by fixed thermometers, and that of the water of condensation at its issue from the coil was similarly shown, while the steam-gauge placed on the boiler gave the other needed data. The calculations are evidently precisely the same as with the preceding type of calorimeter.

The Barrus Calorimeter * (Fig. 2) is essentially of a small surface-condenser. The steam enters by the pipe *j*. The condensing-surface, *a*, is a continuation and enlargement of the supply-pipe, a 1-inch (2.54 cm.) iron pipe with a length of 12 inches (30.4 cm.) of exposed surface. This pipe is under the full pressure of steam. The condensed water collects in the lower parts of the apparatus, where its level is shown in the glass, *e*, and is drawn off by means of the valve, *d*. The injection-water, cooled to a temperature of 40° Fahr., or less, enters the wooden vessel, *o*, through the valve, *b*, and circulates around the condensing pipe, carried downward to the bottom by means of the tube *k*, and overflows at the pipe, *c*, after passing through the mixing chambers, *m*. The amount of water admitted is regulated so as to secure a temperature at the overflow of 75° or 80° Fahr., or the approximate temperature of the surrounding atmosphere. The thermometers, *f* and *g*, which are read to

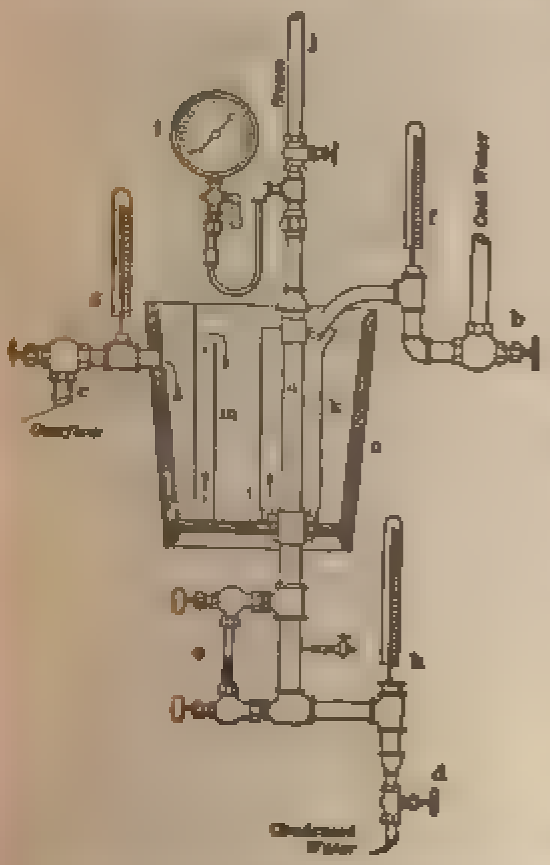

FIG. 2.—THE CONTINUOUS CALORIMETER.

* Trans. Am. Soc. M. E. 1884.

tenths of a degree, show the temperature of injection and overflow water, and the thermometer, *h*, shows that of the condensed water. The overflow water and the condensed water are collected in a system of weighing tanks. The steam-pipe down to the surface of the water, and the pipes in the lower part of the apparatus, are covered with felt.

There is no wire-drawing of the steam, and no allowance to be made for specific heat of the apparatus. The only correction to be made of material amount is for radiation from the pipes covered with felt, and this can be accurately determined by an independent radiation experiment, made when the condenser vessel is empty.

Another form of instrument devised by the same engineer is arranged in such manner as to permit the steam from the boiler to be dried and the quantity of heat so employed measured as a gauge of the amount of water contained in the steam. This form of this apparatus is found very satisfactory.*

The pipe conveying the steam to be tested is usually a half-inch (1.27 cm.) iron pipe. A long thread is cut on this pipe, and it is screwed into the main steam supply-pipe of the boiler in such a manner as to extend diametrically across to the opposite side. The inclosed part is perforated with from 40 to 50 small holes, and the open end of the pipe sealed. If the pipe is screwed into the under side the perforations begin at a distance of one inch (2.54 cm.) from the bottom. The connection is made as short as possible, and covered with felt. Where the calorimeter can be attached to the under side of the main, the distance to the top valve need not exceed six inches (15 cm.). In this position it is self-supporting. The steam for the superheater is also supplied by a half-inch iron pipe, but this may be attached to the main at any convenient point.

Steam to be tested enters by the pipe, which has a jacket. On passing out the thermometer gives its temperature, and it is discharged through a small orifice ⅛ inch (0.32 cm.) in diameter. Steam to be superheated enters and is superheated by a gas-lamp, passes the thermometer,

* Trans. Am. Soc. Mech. Engrs., vol. vii. p. 178.

and issues through an opening like that for the steam. The thermometers are immersed in oil-wells surrounded by the current of steam to be tested, or of that used in drying the boiler steam.

In the operation of this calorimeter steam at full pressure enters the apparatus, and the jacket-steam is heated until a perceptible rise of temperature above that due the pressure indicates that its moisture has been evaporated. The working having become steady, the difference between the temperatures is noted and corrected by deducting the excess above that of moist steam at the observed pressure, and the number of degrees of superheating thus determined, as the rate of flow is the same from both orifices. Here the evaporation of one per cent of moisture from steam at 80 pounds pressure (5.6 kilogs. per sq. cm.) reduces the temperature of superheated steam about 18°.7 Fahr. (10°.4 Cent.), and the percentage of moisture is obtained by dividing the range of superheat, as above, by this number, or generally by the quotient of the latent heat at the observed pressure by 47.5. The following are data and results obtained by the use of this apparatus:

DATA AND RESULTS IN FULL OF CALORIMETER TESTS.

Number for Reference.	Date.	Gauge-pressure	Number of degrees inlet steam was superheated	Number of degrees outlet steam was superheated	Number of degrees wet steam was superheated.	Number of degrees lost by superheated steam due to radiation from calorimeter	Number of degrees representing radiation from supply-pipe	Amount of Moisture in the Wet Steam. Expressed in degrees of superheat	Expressed in percentage.*
1	Apr. 13	89.	99.	54 5	8	8.	9.5	19	1 02
2	" 14	89.	75	37	5.5	8.	9.5	16	0.86
3	" 15	86.	74.	37.	7.	10 5	9.5	10.	0 54
4	" 16	86.	74	39.	9 5	7	9 5	9	0 49
5	" 30	85.	72.	38.	10.5	8.	9 5	6	0 32
6	May 4	80.	77 5	41.5	9 5	6.	9 5	9	0.49
7	" 5	84.	68.	36.5	6.5	7.5	9.5	8.	0.43

NOTE.—The duration of each of these tests was about one hour.

* Obtained by dividing the preceding column by 18.6, the number of degrees corresponding to 1 per cent of moisture.

An exceedingly simple form of calorimeter, practically available when the steam is fairly dry, is that devised by Professor Peabody, which depends on the fact that dry steam is superheated by wire-drawing.*

A piece of pipe six inches in diameter and ten inches long was capped at each end. Into the upper end was fitted a half-inch pipe bringing the steam to be tested, a thermometer cup, and a steam-gauge. From the lower cap an inch pipe led away the exhaust steam. Near the calorimeter was a T which formed a pocket, with a drip at the lower opening, and a branch from the side opening leading to an angle valve in the upper cap of the condenser. The pipe further was well wrapped with hair felt, and the calorimeter was wrapped in asbestos board and hair felt, and covered with russia iron.

Two other calorimeters differ from the first only in size. One is made of a piece of two-inch pipe eight inches long, and the other of a piece of four-inch pipe of the same length. The smaller are more sensitive.

To make an experiment, the valve in the supply-pipe is partly opened, and a valve in the exhaust-pipe is regulated to give the desired pressure in the calorimeter. After the gauge and thermometer attached become steady, their readings are taken, and the reading of the boiler-gauge.

If p_1 is the boiler-pressure, l is the heat of vaporization, and h the heat of the liquid corresponding, x may represent the dry steam in one pound of the mixture from the steam-pipe; $1 - x$ is the water or priming. The heat in one pound of the mixture is

$$xl + h.$$

Let p_2 be the pressure in the calorimeter, and h_1 the total heat, and t_1 the temperature corresponding. Let t_2 be the temperature of the superheated steam by the thermometer. Then the heat in one pound of steam in the calorimeter is

$$h_1 + c_p(t_2 - t_1),$$

* Trans. Am. Soc. M. E., 1888, vol. x.

in which c_p is the specific heat of the superheated steam at constant pressure (0.4808).

Assuming that no heat is lost,

$$xl + h = h_1 + c_p(t_2 - t_1);$$

$$\therefore x = \frac{h_1 + c_p(t_2 - t_1) - h}{l},$$

and the priming is

$$1 - x.$$

The following experiments were made:

Gauge Pressures.		Temperature in the calorimeter F.	Priming.
Boiler.	Calorimeter.		
71.2	38.5	286.7	0.011
60.3	26.8	271.8	0.012
63.0	17.5	264.9	0.013
60.6	7.0	258.8	0.011
69.0	3.7	258.1	0.012

The other calorimeters gave substantially the same results.

This type of calorimeter can be used only when the priming is not excessive; otherwise the wire-drawing will fail to superheat the steam. To find this limit for any pressure, we may assume that the steam is just dry and saturated at that limit in the calorimeter The limit is higher for higher pressures, but the calorimeter can be applied only where the priming is moderate, thus:

Pressure.		Priming.
Absolute.	Gauge.	
300	285.3	0.077
250	235.3	0.070
200	185.3	0.061
175	160.3	0.058
150	135.3	0.052
125	110.3	0.046
100	85.3	0.040
75	60.3	0.032
50	35.3	0.023

The limit may be extended by connecting the exhaust to a condenser. The limit at 100 pounds absolute, with 3 pounds absolute in the calorimeter, thus becomes 0.064, instead of 0.046.

The thermometer should be absolutely reliable. A considerable error in the temperature would produce an inconsiderable effect on the result in other cases. Thus, at 100 pounds absolute with atmospheric pressure in the calorimeter, 10° F. of superheating indicates 0.035 priming, and 15° F. indicates 0.032 priming. A slight error in the gauge-reading has little effect. If reading be apparently 100.5 pounds absolute instead of 100, with 10° of superheating, the priming appears to be 0.033 instead of 0.032.

In the Barrus calorimeter, as has been seen, the steam to be tested is dried and superheated by a stream of highly superheated steam. The following table has been calculated on the assumption that the superheated steam has an initial temperature of 500°, and a final temperature of 10° above the temperature of saturated steam of the given pressure, while the moist steam is supposed to be dried and superheated 5°. The limit under these conditions is widest for lowest pressures, and also is narrower at high pressures than that of the new type:

Pressure.		Priming.
Absolute.	Gauge.	
50	35 3	0 170
75	60.3	0.095
100	185.3	0 086
125	110.3	0.078
150	135.3	0 071
175	160.3	0.065
200	185.3	0.059
250	235.3	0.049
300	285.3	0.040

One or another of these instruments may thus be best, according to pressure of steam carried.

Many other forms of calorimeter have been devised, but space will not permit their description.

47. The Analysis of Gases* issuing from the furnace and passing up the chimney is sometimes an important detail of the work of testing a steam-boiler. Such an investigation involves only an operation of great simplicity which can easily be performed by any engineer. If it is not found convenient to make the analysis in the office of the engineer, he can have the work done, at little expense, by a chemist of known skill and reliability. It is only by a knowledge of the proportions of constituents of the flue-gases that it can be determined whether the combustion is complete, whether the products of combustion are diluted with excess of air, and whether the fuel used has been so burned as to give its best effect. Such analyses also enable the engineer to ascertain the best method of burning the fuel.

In sampling the gases, a matter in regard to which some precaution is advisable, the method of Mr. Hoadly is found very satisfactory.†

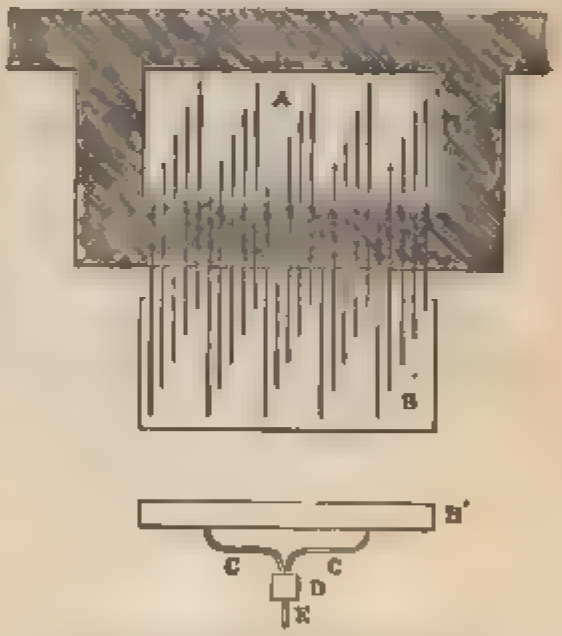

FIG. 3.—FLUE-GAS SAMPLING.

Very great diversities in composition often exist in the same flue at the same time. To obtain a sample, allow one orifice to draw off gases through for each 25 sq. inches (161 sq. cm.) of cross-section of flue. The pipes must be of equal diameter and of equal length. These should be secured in a box of galvanized sheet-iron, equal in thickness to one course of brick, so that the ends may be evenly distributed over the flue *A* (Fig. 3), and their other open ends inclosed in the

* Consult Handbook of Gas Analysis, by C. Winkler. London: J. Van Voorst. 1885.

† Trans. Am. Soc. M. E., vol. vi.

receiver *B*. If the flue gases be drawn off from the receiver *B* by four tubes *C C*, into a mixing box *D*, beneath, a good mixture can be obtained.

The sampling of the gas should be carried out at intervals of 10 to 15 minutes throughout the trial. The gas should be received in an air-tight pipe or jar. The composition of the gases should be determined as far as regards carbonic acid, carbonic oxide, and oxygen. The tube should be of porcelain or glass for very hot flues, since iron tubes at such temperatures are oxidized. Supposing an analysis of the gas give K per cent of carbonic acid, O per cent of oxygen, and N per cent of nitrogen, then the proportion of air actually used to the theoretical quantity required is 1 to x.

Where

$$x = \frac{N}{N - \frac{79}{21}O} \text{ or } \frac{21}{21 - 79\frac{O}{N}}.$$

unity of weight of this coal will then give, at a temperature of 0° and a pressure of one atmosphere,

$$\frac{1854}{10}C = \text{carbonic acid};$$

$$\frac{KO}{K} = \text{oxygen};$$

$$\frac{KN}{K} = \text{nitrogen}.$$

The quantity of moisture in the escaping gases may be calculated from the moisture in the coal, from that formed by burning the hydrogen, and from that contained in the air admitted to the furnace where the latter has been determined. Any serious break in the setting can be detected by filling the grate with smoky coal and then closing the damper.

The following sketch shows the apparatus employed by Mr. Wilson.*

* Journal Society of Arts, Feb. 1889.

A. Apparatus employed for the gas analyses. The whole apparatus being filled with mercury, the gas is introduced into the eudiometer *a* and its volume measured. The stopcock *b* and the three-way cock *c* are then opened, and the gas passes over into the laboratory vessel *d*, followed by some mercury to drive all the gas out of the capillary tube. The reagent is then poured into the cup *e*, and admitted to the laboratory vessel by the three-way cock. When the absorption is complete, the mercury bottle is placed on the upper shelf and the cocks being opened the gas passes back into the eudiometer. When the reagent rises to *c* the three-way cock is turned to communicate with the cup so that the reagent passes into it. Some mercury is then driven over into the eudiometer to clear the gas from the capillary tube, and the volume is again read.

The two ends of the capillary tubes at *f* are made funnel-shaped, and connected by a thick india-rubber tube. By lowering the eudiometer a little when the gas is passed from *a* to *d*, and raising it for the passage in the opposite direction, the whole of the gas is driven out of the tube.

FIG. 4.—APPARATUS FOR GAS ANALYSIS.

B. One of the tubes used for taking samples of gas. The sampler, completely filled with mercury, is connected with the gas to be taken by means of an india-rubber tube (previously aspirated if necessary). The vessel is then inclined so as to allow mercury to flow out of the opposite tube until only enough remains to seal up the sample.

The apparatus designed by Professor Elliott, and employed in work carried on under the direction of the Author, consists,

as shown in Fig. 5, of two vertical glass tubes, *AB*, *A'B'*, joined by rubber-tubing, *E*, at their upper ends. The large tube, *AB*, is the treating, the smaller, *A'B'*, the measuring tube; the latter is suitably graduated to cubic centimetres. Water-bottles, *K*, *L*, are connected with the lower ends of the tubes by tubing, *NO*, *N'O'*, and are used in effecting transfer of the gas from tube to tube. *M* is a funnel through which the reagents used may be introduced. *G*, *F*, and *I* are cocks of suitable size and construction.

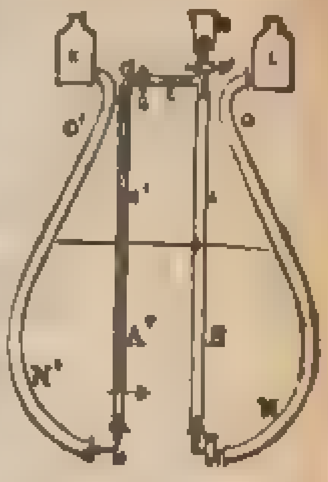

FIG. 5. APPARATUS FOR GAS ANALYSIS.

In filling the apparatus it is set up conveniently near the flue, and the line of tubing from the collector, within the latter, is connected with the tube *AB*. The receiver *L* being detached the lower end of *AB* is connected with an aspirator or equivalent apparatus, such, for example, as might be improvised by the use of an air-tight tank or a barrel; and the flow thus produced, when the aspirator is emptied of its water, fills the tube *AB* with gas drawn from the flue. It is retained by closing the valves *F* and *I*, which had been open during the operation of filling. The tube is then disconnected from the aspirator, and the receiver, or bottle, *L*, connected as shown, and in such manner that no air can reach the tube *AB*.

Removing the apparatus to the laboratory or other convenient location, the analysis is made as follows:

Pass into *A'B'* a convenient volume, as 100 c.c. of the gas, and discharge the remainder through the valve and funnel *F* and *M*, filling the tube *AB* with water from *L*. Transfer the measured gas back to *AB*, through *E*, and add a solution from *M*, which will absorb some one constituent. Return the gas to *A'B'*, and again read its volumes. The difference is the quantity of gas absorbed. Repeat this process, using next an absorbent which will take up a second constituent of the gas, and thus obtain a second measure of volume; and thus continue until all the desired determinations are made. All readings should be made at the same temperature, or practically so. The tube

AB should be well washed at each operation, in order that no reagent should be affected by traces of that previously used.

The absorbents employed are best taken in the following order:

1. Caustic potash—to absorb carbonic acid.
2. Potassium pyrogallate—to absorb free oxygen.
3. Cuprous chloride in concentrated hydrochloric-acid solution—to absorb carbonic oxide.

After their use nitrogen will remain, and will be measured as a balance which, added to the sum of the measured volumes of gases absorbed, should give the original total. Where weights are to be determined, the volumetric measures obtained as above are to be reduced by the usual process.

The atomic weights of the principal constituents being, oxygen, 16; nitrogen, 14; carbon monoxide, 28; carbon dioxide, 44, we shall have by percentages, where the symbols represent per cent in volumes, for each, when the total is

$$M = 14N + 16O + 28CO + 44CO_2,$$

$$\frac{14N}{M}, \quad \frac{16O}{M}, \quad \frac{28CO}{M}, \quad \frac{44CO_2}{M}, \text{ respectively.}$$

Since the total per cent of oxygen is measured by $\frac{32}{44}CO_2 + \frac{16}{28}CO +$ free oxygen, and the total per cent of carbon is $\frac{12}{44}CO_2 + \frac{12}{28}CO$, we shall have for the percentage of each,

$$O' = \frac{32 \times 44 \times CO_2}{44M} + \frac{16 \times 28 \times CO}{28M} + \frac{16O}{M};$$

$$C' = \frac{12 \times 44 \times CO_2}{44M} + \frac{12 \times 28 \times CO}{28M};$$

or,

$$O' = 32\frac{CO_2}{M} + 16\frac{(CO + O)}{M};$$

$$C' = 12\frac{CO_2}{M} + 12\frac{CO}{M}.$$

The total oxygen is that which entered the furnace as the supporter of combustion, and is a measure of the air supplied. The ratio of free to combined oxygen is a measure of the ratio of the air acting as a diluent simply to that supporting combustion.

Thus these measurements exhibit the efficiency of combustion, the quantity of air employed, and the magnitude of the wastes of heat at the chimney, occurring through imperfect combustion or excess of air-supply. It is evident, however, that where moisture or steam accompanies the gases, it escapes measurement; this, however, introduces no important error in ordinary work.

Efficiency of combustion is indicated by the analysis of the flue-gases with very great certainty. The appearance of carbon monoxide at the chimney proves the combustion to be imperfect in proportion as it is more or less abundant. The presence of unconsumed oxygen, on the other hand, in the absence of carbon monoxide, proves an excess of air-supply. Both gases appearing is a proof of incomplete intermixture of air and combustible, or of so low a temperature of furnace as to check combustion. This analysis being compared with that of the fuel reveals the character and the perfection of combustion, and permits a very exact determination to be made of the specific heat of the gases, and is thus a check on calculations of wasted heat. (See Appendix XVI.)

48. Draught-gauges are made for the purpose of determining the head-producing draught and the intensity of the draught, which are of many forms, but which usually depend upon the measurement of the head of water which balances that head at the chimney. A very compact and accurate form

of draught-gauge, used by the Author with very satisfactory results, is that of Mr. J. M. Allen (Fig. 6).

A and *A'* are glass tubes, mounted as shown, communicating with each other by a passage through the base, which may be closed by means of the stop-cock shown. Surrounding the glass tubes are two brass rings, *B* and *B'*. These rings are attached to blocks which slide in dovetailed grooves in the

Fig. 6.—Draught-gauge.

body of the instrument, and may be moved up and down by screws at *F F'*. The scales are divided into fortieths of an inch, and read to thousandths of an inch by the verniers *e* and *e'*, which are attached to the sliding rings *B B'*. If the two short rings are set at different heights, the difference in readings will give the difference of level. The thermometer is for the purpose of noting the temperature of the external air. The method of using the instrument is as follows:* At a con-

* *The Locomotive*, May, 1884, p. 67.

venient point near the base of the chimney a hole is made large enough to insert a thermometer. The height from this opening to the top of chimney, and also of grates, should be noted. The chimney-gauge is attached to some convenient wall. The tubes are filled about half full of water, when the verniers afford an easy means of setting it perpendicular. One end of a flexible rubber tube is then inserted into the upper end of one of the glass tubes, and the other end of the tube is in the chimney-flue. The tubes *B B′* are adjusted until their upper ends are just tangent to the surface of the water in the two tubes. The reading of the two scales is then taken, and their difference. At the same time the temperature of the flue is noted, as well as that of the external atmosphere. Comparison may then be made with the following table, computed for use in this connection for a chimney 100 feet high, with various temperatures outside and inside of the flue, and on the supposition that the *temperature of the chimney is uniform from top to bottom*—an inaccurate though usual assumption, however. For other heights than 100 feet, the theoretical height is found by simple proportion, thus: Suppose the external temperature is 60°, temperature of flue 380°, height of chimney 137 feet, then under 60° at the top of the table, and opposite to 380° interpolated in the left-hand margin, we find .52″.

Then 100 : 137 :: .52″ : .71″, which is the required height for a 137-foot chimney, and similarly for any other height.

HEIGHT OF WATER COLUMN DUE TO UNBALANCED PRESSURE IN CHIMNEY 100 FEET HIGH.

Temperature in the Chimney Fahr.	Temperature (Fahr.) of the External Air—Barometer, 14.7				
	20°	40°	60°	80°	100°
220	.419	.355	.298	.244	.192
250	.468	.405	.347	.294	.242
300	.541	.478	.420	.367	.315
350	.607	.543	.486	.432	.380
400	.662	598	541	.488	.436
450	714	.651	.593	.540	.488
500	.760	697	.639	586	.534

The most common form of gauge-testing apparatus is shown in the accompanying engraving. The standard gauge, which is known by comparison with a mercury column or by other

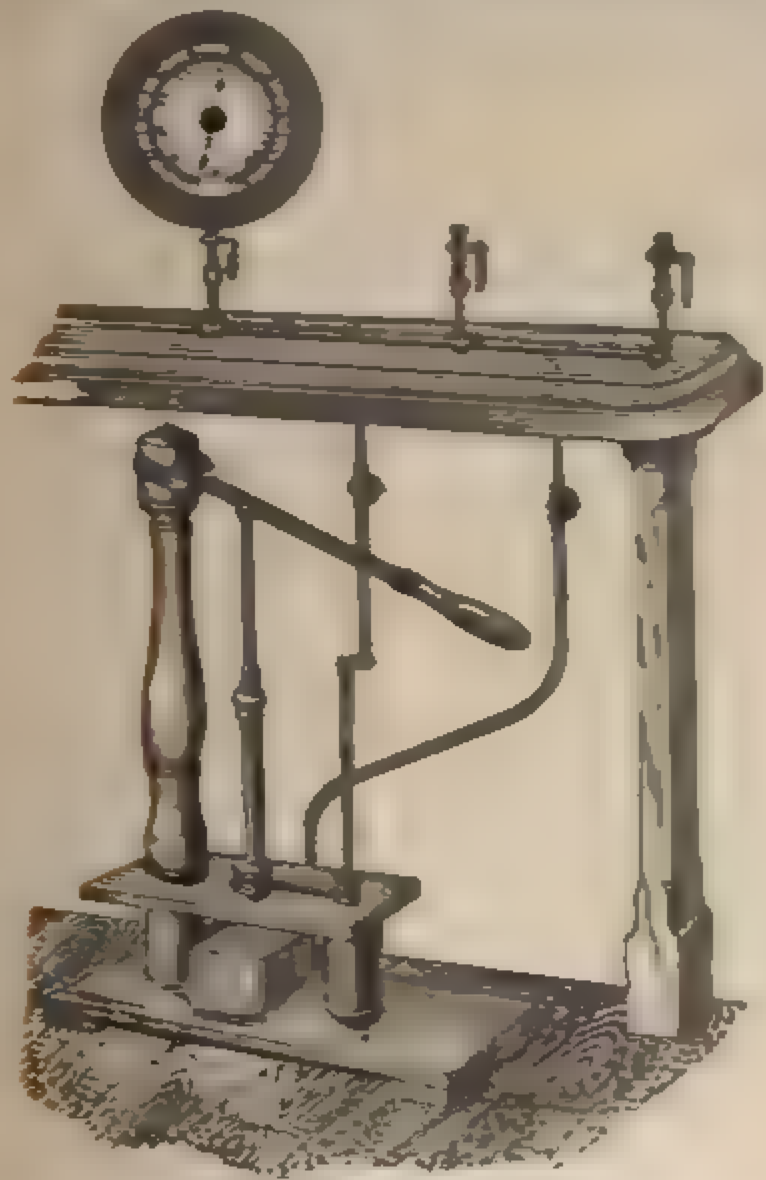

FIG. 7. GAUGE-TESTING PUMP.

test to be right, is mounted as shown. The instrument to be tested is attached to one of the other cocks, and, both being subjected to the same pressures, a comparison of their readings will exhibit the errors of the second gauge.

49. A Sample Trial is described in the following report, which will illustrate well the methods and results of a carefully made test of a boiler in which a complete trial was attempted, under the direction of the Author.*

* Sci. Am. Supplement, No. 641, p. 10234.

Trial of a Water-tube Boiler.

This boiler was used to supply steam to one or more engines, as needed, or to heat the buildings of the college. The principal dimensions are as follows:

Length of drum,	13 ft.
Diameter,	2 ft. 6 in.
Number of water-tubes,	40
Outside diameter of tubes,	4 in.
Length,	13 ft. 8 in.
Width of furnace,	3 ft. 3½ in.
Length of furnace,	6 ft. 1 in.
Length of grate-bars,	3 ft.
Width of grate-bars,	¾ in.
Width of air-spaces,	¾ in.
Number of grate-bars,	54
Area of chimney,	3.65 sq. ft.
Height of chimney,	60.25 ft.
Area of grate-surface,	20 sq. ft.
Area of heating-surface,	682.57 sq. ft.
Area for draught between tubes,	4.75 "
Ratio of grate to heating-surface,	1 : 34.1
Ratio of draught area to grate,	0.25
Ratio of grate-surface to cross-section of chimney,	5.48
Ratio of area of grate to area of air-spaces,	2.24
Whole area of damper opening,	3 sq. ft.

The main steam-pipe after passing horizontally to the rear of the "setting" descends vertically a distance of 4 ft. and passes out of the boiler-room to the chimney. Draught is produced by a chimney which rises directly at the back end of the boiler, the first 9½ ft. being brick and the remainder a sheet-iron cylindrical stack. A vertical sliding damper is placed in the opening leading to the chimney. Two partitions of fire-

brick supported by iron plates are placed transversely across the nest of water-tubes. The first is 7 ft. 1 in. from the front end of the tubes, and the second 3 ft. 7 in. from the first. These partitions cause the gases to pass among the tubes three times, then across the rising tubes into the back connection, and from there to the chimney.

The object of the trial was: 1. To determine the evaporative efficiency of the boiler. 2. To estimate the horse-power developed under ordinary working conditions, a horse-power being taken as equivalent to 30 lbs. of feed-water supplied per hour at a temperature of 100° F., and evaporated under 70 lbs. gauge-pressure.

FIG. 8.—GAS SAMPLER.

Previous to the test all cracks and holes in the setting and around the doors leading to the flues were carefully stopped with fire-clay and mortar. The blow-off and return "drip" pipes were disconnected and caps placed on the exposed ends. An injector feed-pipe connected with the boiler was left in place, as its disconnection would be attended with some difficulty. The overflow-pipe was, however, left open, in order to detect any leak which might occur. The feed-pipe was disconnected from the "mains," and a suction-pipe from it placed in a barrel into which the feed water was run after having been weighed. A pipe leading to the outside of the boiler-house was connected with the main steam-pipe, so that all steam made by the boiler, over and above that required to run the engine and heat the buildings, could be discharged into the air.

At 7 A.M. April 28, the fire, which had been banked on the preceding evening, was started, and the steam pressure brought to 80 lbs. by the large gauge. The fire was then quickly drawn and the contents of the ash-pit removed. A new fire was started immediately with a weighed quantity of hemlock wood and brought to the normal condition with coal. The amount of water shown by the water-glass was noted. At 8 A.M. the engine was started, and the trial commenced. Both ash-pit doors

were left open at first and the damper wide open. The damper was lowered 3 in. at 9.30 A.M., and at 12.50 a further amount of 3 in. At 11.17 A.M. one of the ash-pit doors was closed and so remained during the remainder of the trial. The effect of this arrangement of damper and draught door was observed in the higher temperature of the flue gases at the base of the chimney.

The fuel used was anthracite coal, known in the market as "grate coal." An average sample of this coal was weighed, pulverized, and placed in an evaporating oven to dry. After seven hours it was found to have lost 3.81 per cent. in weight. In working up the results of the trial, this figure was taken to represent the percentage of moisture in the coal. The coal was weighed by the barrow load in uniform charges of 200 lbs. each, and dumped before the door as needed.

FIG. 9.—PYROMETER.

The stoking or firing was performed regularly every half hour and fire cleaned every third time. During the period of stoking, the back damper was closed to avoid loss of heat by the current of cold air which otherwise would rush through the heated flues. The feed-water was drawn from the mains into a barrel placed on a platform scale, where it was carefully weighed. It was then drawn off into another barrel, from which it was pumped into the boiler by a steam-pump of the ordinary type. It was the endeavor to deliver the water to the boiler as continuously as possible The temperature of the feed-water was noted at each weighing.

The observations which were called for, and the results of which were finally recorded as the log of the trial, were such as would ordinarily be demanded in any usual case of engineering practice of this character, and were sufficient to enable the observers to make all essential computations; while none were made which were not either of importance in that respect, or of real interest to the engineer concerned in the questions proposed to be settled by the trial as conducted.

The method of conduct of the trial, in tolerably full detail, is described in the succeeding pages. The observations made are indicated; the processes involved in their reduction, computation, and tabulation are exhibited and illustrated; and the final deductions and conclusions are stated at length.

The following observations were made every half-hour:

1. Temperature of flue gases at the base of chimney.
2. Temperature of boiler-room.
3. Temperature of outside air.
4. Reading of draught pressure-gauge.
5. Readings of the several steam-gauges.
6. The pyrometer used in measuring the temperature of the flue gases had previously been compared with a mercury thermometer between temperatures of 213° F. and 322° F. This was accomplished by means of a simple apparatus shown in Fig. 9. The stem of the pyrometer was inclosed in a steam-pipe which has communication to the boiler through a smaller pipe fitted with a stop-valve. The thermometer used in the comparison was also screwed into the larger pipe. As steam was admitted the mercury rose, and soon registered a temperature corresponding to the steam-pressure, which was kept constant for several minutes until the pyrometer reading no longer changed.

Both readings were noted, and more steam admitted, giving a higher temperature.

The several readings were "plotted," and the law of variation of the pyrometer from the thermometer reading was found to be approximately a straight line, continually falling below and diverging from the line representing the temperature as read from the thermometer.

The pyrometer was corrected from this line, and is believed to be approximately correct.

The draught pressure-gauge, which was attached to the stack near the base, was made for the Sibley College laboratories by the Hartford Steam Boiler Insurance Co. It consisted of a U-tube partially filled with water and provided with a movable vernier and scale for measuring the difference in level of the water in the two arms.

A recording steam-gauge and a mercury-gauge were attached to the boiler in addition to the large gauge ordinarily used.

The mercury-gauge was taken as the standard, and the others corrected by it.

Experiments were made every hour to determine the quality of the steam. A well-made barrel which had been thoroughly shellacked inside was placed on a very sensitive standardized platform scale, made for this work, the beam of which was graduated to $\frac{1}{20}$ of a pound and provided with a sliding poise.

The steam was taken from the main steam-pipe, 1 ft. from its connection with the boiler, and was conducted to the calorimeter through a $\frac{1}{2}$-in. pipe, 9 ft. long, to the end of which was attached a piece of rubber hose 7 ft. long. The pipe was covered with hair-felt to prevent radiation of heat. Before placing the end of the hose in the calorimeter, steam was allowed to blow through until all the water of condensation had been discharged and the pipe and hose were thoroughly warmed up. The end of the hose was given an inclination downward toward the bottom of the barrel by means of a light strip of wood fastened to it. The steam passing into the condensing water at an angle produced a strong agitation, and thus a thorough mixture of the water was effected.* A standard Centigrade thermometer, graduated to tenths of one degree, was used with the calorimeter, and the readings were afterward reduced to the Fahrenheit scale.

During the trial five samples of flue gas were taken for analysis. The tabulated results of the analysis are as follows:

PER CENT. BY VOLUME.

No.	Time.	CO_2, Observed.	Free O, Observed.	CO, Calculated	N, Calculated
1	8.30 A.M.	12	5 2	4.6	78.13
2	10.20 A.M.	12	6.7	2.16	79.13
3	12.20 P M	11 1	7 9	1 6	79 3
4	2.20 P M.	11.7	6.8	2.5	79
5	4.20 P.M.	11.5	7	2 5	79

* Probably, on the whole, as good an arrangement as any plan involving the use of stirring apparatus. See Manual of the Steam Boiler.

BY WEIGHT.

No.	Time.	CO_2, Calculated.	Free O, Calculated.	CO, Calculated.	N, Calculated.
1	8.30 A.M.	17.56	5.5	4.33	72.62
2	10.20 A.M.	17.52	7.07	2	73.40
3	12.20 P.M.	16.27	8.39	1.95	73.86
4	2.20 P.M.	17.11	7.19	2.32	73.38
5	4.20 P.M.	16.83	7.41	2.32	73.44

No.	Time.	Per Cent by Weight, Total O.	Per Cent by Weight, Total C.	Air Supplied, Per lb. C.	$\frac{\text{Free O}}{\text{Combined O}}$
1	8.30 A.M.	20.74	6.65	14	0.36
2	10.20 A.M.	20.96	5.64	16.7	0.51
3	12.20 P.M.	21.31	5.27	18	0.64
4	2.20 P.M.	20.95	5.66	16.6	0.52
5	4.20 P.M.	20.97	5.58	16.9	0.54

Professor Elliott's apparatus, Fig. 10, was used for the analysis. For the absorption of CO_2 a solution of potassic hydrate (1 to 20) was used, and for oxygen absorption, potassic pyrogallate; this latter being prepared by adding 5 per cent of pyrogallic acid to a solution of potassic hydrate (1 to 8). Numbers 1 and 2 were tested for CO with cuprous chloride, but as none was absorbed, and it was evidently present, the amount was calculated as follows:

For No. 1 we have 12 per cent CO_2, whose volume is equal to the volume of the O which combined to form it, and 52 per cent of free O. The volume of O in these two is, therefore, $= 12 + 5.2 = 17.2$ per cent. Assuming that the atmospheric air is composed of 4 parts of N and 1 part of O, by volume, to correspond to this 17.2 per cent of O we should have $17.2 \times 4 = 68.8$ per cent N; but after absorbing the 17.2 per cent of CO_2 and O, there remains $100 - 17.2 = 82.8$ per cent. Taking 68.8 per cent from 82.8 per cent, we have 14 per cent, which must be composed of N and CO. Since the volume of CO is equal to twice the volume of the combined O, we shall have the volume of $O = \frac{CO}{2}$, and since there is four times as

much N as O, the $N = \frac{4CO}{2} = 2CO$. Therefore of this 14 per cent. 1 part is CO and 2 parts are N; $\therefore CO = \frac{14}{3} = 4.6+$, and $N = 4.6 \times 2 = 9.3+$, which being added to the 68.8 per cent. N, which corresponds to the free O, and that of the CO_2

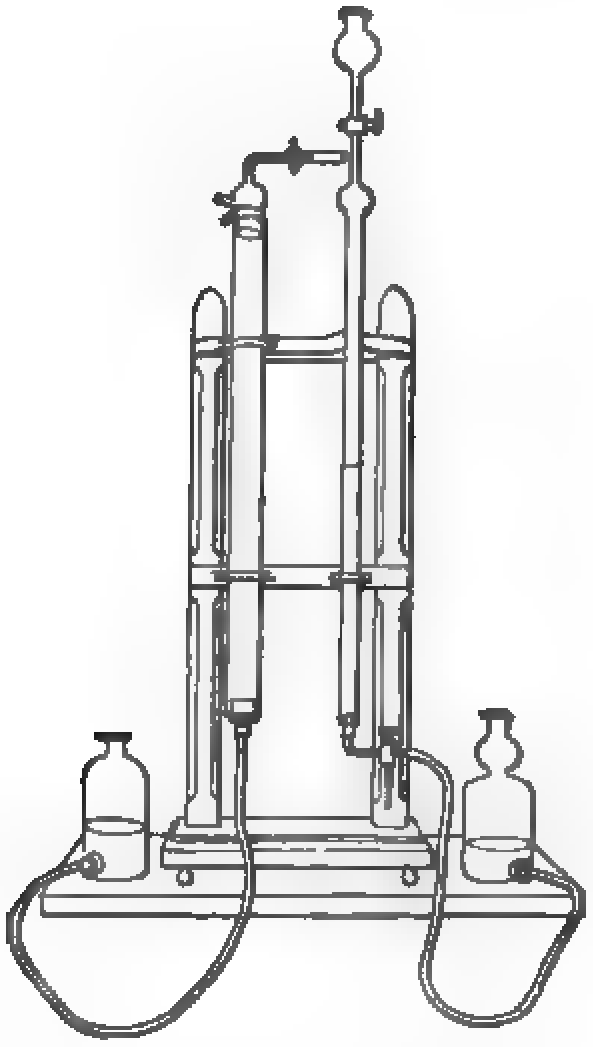

FIG. 10.—GAS ANALYSIS.

= 78.13 per cent. To reduce per cent. by volume to per cent. by weight, we use the following constants:

Weight of	1	liter of	CO_2,	1.9774	grams.
"	" 1	"	" O,	1.43	"
"	" 1	"	" CO,	1.254	"
"	" 1	"	" N,	1.256	"

Multiplying the per cent. by the volume of each gas by the weight of a liter of that gas, we get certain values, a, a', a'', etc. Taking the sum of these $= s$, then the per cent. of weight would be $\frac{a}{s}$, $\frac{a'}{s}$, $\frac{a''}{s}$, etc.

To get the total O and the total C: The atomic weight of O $= 16$ and of C $= 12$; $\therefore$ the amount of O in $CO_2 = \frac{2 \times 16}{12 + (2 \times 16)} = \frac{8}{11}$ and the amount of C $= \frac{12}{12 + (2 \times 16)} = \frac{12}{44} = \frac{3}{11}$.

In the same manner the amount of O in CO $= \frac{16}{12 + 16} = \frac{4}{7}$, and C $= \frac{3}{7}$.

Hence the total O $= \frac{8}{11} CO_2 + \frac{4}{7} CO + O$, and the total C $= \frac{3}{11} CO_2 + \frac{3}{7} CO$.

To get the ratio of air for dilution to that for combustion, we have $\frac{\text{Free O}}{\text{Combined O}} = \frac{O}{\frac{8}{11} CO_2 + \frac{4}{7} CO}$.

To get a measure of the air supplied per pound of carbon, we take the per cent. by weight of total O + per cent. by weight of N, and $\div$ by the per cent. by weight of C.

At the time of taking the samples of gas the conditions were as follows:

No. 1. Fire had not completely burned clear from first firing. Back damper was wide open, as were the draught doors.

No. 2. Fire burning clear. Back damper dropped 3 inches. Draught doors wide open.

No. 3. Fire clear. Back damper 3 inches down. One draught door closed.

No. 4. Fires clear. Back damper 6 inches down. One draught door closed.

No. 5. Same as No. 4.

From these figures the following results are obtained:

FLUE GASES.

Average free O,	by weight,	7.108	per cent.
" " CO_2,	" "	17.059	"
" " CO,	" "	2.584	"
" " N,	" "	73.34	"

The average ratio of the amount of air for *dilution* of the gaseous products of combustion to that necessary for combustion is as 0.514 to 1, *i.e.*, 16.44 lb. of air per pound of combustible, or 1.37 times the theoretical amount. The ratio of amount of air reqired for the dilution of the gaseous products of combustion to that necessary for combustion is variously estimated by different authors, but is generally taken as $\frac{1}{2}:1$. It will be seen that a very small per cent. of CO passed up the chimney, the average being 2.67 per cent. by volume, showing the combustion to be nearly complete.

The waste by air in the chimney is calculated by the following formula:

Let W = the number of pounds of air for combustion and dilution;
t = temperature of chimney;
t' = temperature of external air;
S = specific heat of air.

Then

$$H = W(t - t')S,$$

where H is the number of heat-units carried off by the escaping gases.

We have

$$W = 16.44;$$
$$t = 435.7^\circ \text{ Fahr.};$$
$$t' = 60.39^\circ \text{ ״}$$
$$S = 0.238.$$

Hence

$$H = 16.44\,(435.7 - 60.39)\,0.238 = 1468.48 \text{ units.}$$

Assuming that a pound of coal will evaporate 15 pounds of water from and at 212° Fahr., or equal to 14,491 heat-units, the loss by chimney is 0.101.

The height of chimney required under the above conditions is found from Rankine's formulæ as follows:

Let W = weight of fuel burned in the furnace, per second;
V_0 = the volume at 32° F. of air supplied per lb. of fuel;
T = the absolute temperature of gas discharged by the chimney;
A = sectional area of damper opening.

Then the velocity of the current in the chimney in feet per second is

$$u = \frac{WV_0T}{AT_0}.$$

Hence

$$u = \frac{0.06869\,(12.386 \times 16.45)\,896.9}{2.222 \times 493} = 11.449 \text{ ft. per sec.},$$

and h, the head required to produce this draught, is

$$h = \frac{u^2}{2g}\left(1 + G + \frac{fl}{m}\right),$$

where

l = the whole length of chimney and flue leading to it in feet;

m = its hydraulic mean radius;

f = coefficient of friction; (estimated by Pectet at 0.012;)

G = a factor of resistance for the passage of air through grate and fuel; (given by Pectet as 12.)

Hence

$$h = \frac{(11.449)^2}{2 \times 32.2}\left(13 + \frac{0.012 \times 93}{534}\right) = 30.7138 \text{ ft.}$$

Then

$$H = h \div (0.96 \frac{T_1}{T_2} - 1),$$

where H is the height of chimney:

$$H = 30.7138 \div (0.96 \frac{896.9}{521.59} - 1) = 47.25 \text{ ft.}$$

The actual height as measured was 60.25 ft. The difference between this and the calculated height, or the throttling effect of the damper, being

$$60.25 - 47.25 = 13 \text{ ft.}$$

The following data were taken during the trial:

Total coal, ...	2963.2 lbs.
Total ash and waste, ...	342 "
Per cent. ash and waste, ...	11.5 "

The wood used was considered as equal to 0.4 the same weight of coal.

At 6 P.M. the fire was hauled and the unconsumed coal and the contents of the ash-pit were weighed up dry. The height of water in the gauge-glass was brought to the same position

as at the start, and all conditions made as near those at the beginning of the trial as possible.

The following are the records:

Total weight of water, ...	23, 912.5	lbs.
Average temperature, . . .	46.125	Fahr.

AVERAGE PRESSURES.

Mercury-gauge, ...	85.78	lbs.
Edson gauge, by record chart, corrected,	85.4	"

Barometer readings were taken from the report of the University Signal Service.

Let x = weight of dry steam run into calorimeter;
y' = weight of water in the steam;
y = percentage of priming;
W = weight of condensing water;
w = weight of condensed steam;
t'' = the initial temperature;
t' = the final temperature;
T = heat-units per lb. of steam;
t = heat-units per lb. of water.

Then

Range of temperature, . . . $R = t' - t''$;
Heat transferred to calorimeter, $U = WxR$;
Heat from steam, per lb., . . $H = T - t'$;
Heat from water, per lb., . . $h = t - t'$.

$$x + y' = w, \quad \ldots \ldots \ldots \quad (1)$$

$$Hx + hy' = U. \quad \ldots \ldots \ldots \quad (2)$$

From 1 and 2,

$$x(H - h) = U - wh,$$

$$x = \frac{U - wh}{H - h}.$$

Percentage of priming,

$$y = 100 \frac{w - x}{w}.$$

The ten calorimeter experiments gave the following average results:

Steam-pressure,	100.522	lbs.
Weight of condensing water,	382.985	"
Weight of steam condensed,	24.335	"
Initial temperature,	47.979	Fahr.
Final temperature,	117.430	"
Range of temperature,	69.45	"
Dry steam run into the calorimeter,	24.2853	lbs.
Per cent. of priming,	0.189	

DATA AND RESULTS.

Date of test,	April 28, 1887	
Weight of wood used in lighting fires,	245.5	lbs.
Equivalent value of wood referred to fuel,	98.2	"
Weight of anthracite coal used,	3265	"
Total weight of fuel,	3363.2	"
Weight of unconsumed coal left on the grates,	400	"
Total weight of fuel consumed,	2963.2	"
Weight of ashes and clinkers,	342	"
Percentage of ash and clinkers to fuel consumed,	11.5	
Percentage of moisture in coal,	3.81	
Weight of fuel, less moisture,	2752.11	"
Weight of combustible used,	2410.11	"
Total weight of feed-water supplied and evaporated,	23912.5	"
Average steam-pressure,	85.4	"
Average temperature feed-water,	46.71	Fr.
Average temperature of escaping gases,	435.7	"
Average force of draught in inches of water,	0.275	in.
Water evaporated per lb. of fuel, observed conditions,	8.68	lbs.

Equivalent evaporation, per lb. of fuel, from and at 212° Fahr.,	10.486 lbs.
Water evaporated per lb. of combustible,	9.92 "
Equivalent from and at 212°,	11.984 "
Average temperature of boiler-room,	80.06 Fr.
Average temperature of outside air,	60.39 "
Average height of barometer,	28.702 in.
Horse-power developed on a basis of 30 lbs. of feed-water supplied at 100° Fahr. and evaporated at 70 lbs.,	83.75
Rated horse-power,	61
Per cent above rated capacity,	37

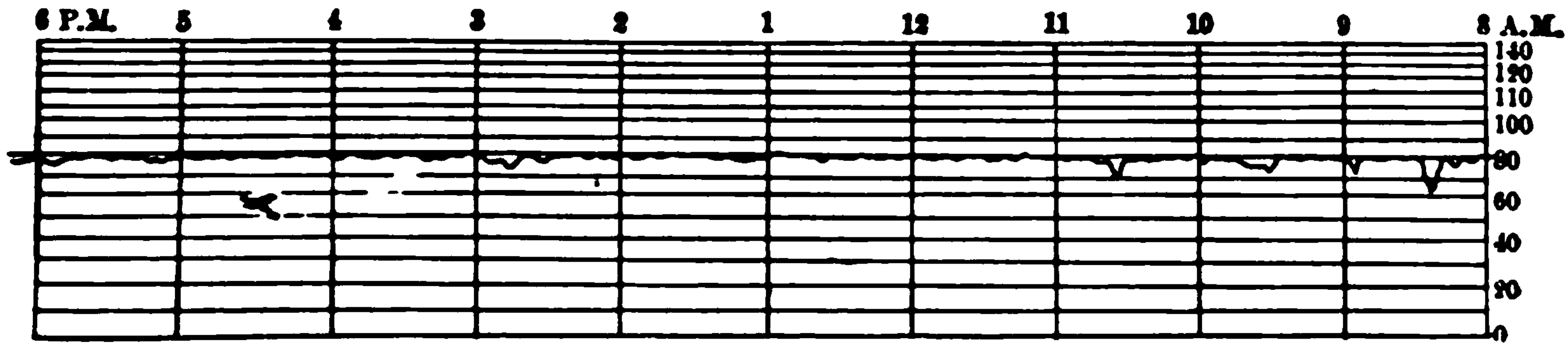

FIG. 11.—Autographic record of steam-pressure during the trial, from Edson gauge.
Mean pressure as shown on the diagram, 78.4 lbs. per sq. in.
Mean pressure, corrected, ... 85.4 " " "

CHAPTER IV.

THE STEAM-ENGINE INDICATOR.

50. The Indicator and the Dynamometer are the instruments employed in the engine-test proper. The purpose of their use is the measurement by the one of all fluctuations of pressure and of volume of the steam within the working cylinder, and of the work done and power developed by its action on the piston, the gross work performed by the transformation of heat-energy, and by the other the net work of the engine, the work done and power available at the engine-shaft for useful application. The difference between these two quantities is the measure of the lost energy and the wasted power, due to the resistances of the machine itself, the sum of the friction-resistances and the back pressure on the exhaust side of the piston, if the gross indicated power is measured to the line of external atmospheric or condenser pressure, or to friction alone if the power is taken as exclusive of back-pressure work.

The indicator is sometimes a "continuous indicator," giving a running, continuous, record of power developed. The most usual form, however, is that which gives a graphical representation of the whole cycle on one side of the piston, and thus permits a study to be made of all the variations of pressure throughout the stroke, and thus a deduction of the conditions of valve adjustment or setting, and of its action in distributing steam. The dynamometer is sometimes of the transmitting form, stationed between the engine and its work or any introduced resistance; but it is most usually of the type known as the Prony brake or the absorbing dynamometer, and takes up the whole external power of the engine, converting

all that energy into heat; which heat it wastes by conduction and radiation to surrounding objects or to a stream of water kept flowing over it, or through the rim of the brake-wheel.

51. The Principles of the Indicator of the usual construction and type, those which govern its action and determine its value, are as follows:

(1) It must exhibit with precision the pressure of the steam within the working cylinder at every instant throughout the stroke.

(2) Simultaneous measures must be given of the position of the piston corresponding to the given pressure, each instant.

(3) The diagram produced must be so made, automatically, as to have its ordinates exactly proportional to the steam-pressures and its abscissas as accurately proportional to the motions of the piston, each point in the curve, by its coördinates, giving a measure, simultaneously, of these two quantities.

(4) The diagram must be unaffected either by the forces acting on the engine, other than that which it is constructed to measure, or those brought into existence by its own motions, and whether they are active or passive, whether of inertia or of friction. The ideal indicator would be an instrument possessing the above qualities, and would trace a conveniently large diagram with absolute exactness. It would be free from inertia and perfectly inflexible in every part. As these ideal conditions are approximated, differences among the best makes of indicators become less and less, and should finally disappear. As they are now sometimes made, however, unless carefully selected and as carefully tested and standardized, it is perfectly possible for differences of very considerable importance to be observed. The Author has sometimes noted results from indicators, simultaneously used, varying from ten to fifteen per cent.

52. The Essentials of a Good Indicator are:

(1) Such form and construction as will insure its meeting the prescribed general conditions—accuracy of representation of the variations of steam-pressure and the simultaneous movement of the piston at all times.

(2) Such simplicity of form as will make it free from liability to accident and failure in operation.

(3) Such lightness of parts, and such rigidity as a whole, as will prevent any inaccuracy of indications arising from its inertia.

(4) It should be easily, conveniently, and safely attachable and removable, and readily and handily manipulated.

Stiffness, lightness, and exactness of standardization are the prime essentials. The springs should be exactly standard; the moving parts as light as is consistent with proper strength and stiffness; the stationary parts should be carefully proportioned and rigid; the whole instrument should be portable, and yet the scale of its diagram as large as practicable, and consistent with exactness in its production.

53. The Forms of the Indicator, as commonly constructed, are usually very similar, the more important differences being found in the recording system. The original indicator employed from about 1814* by Watt, Fig. 12, consisted of a small steam-cylinder, *AA*, traversed by a piston, *K*, the latter held by a spring, *F*, which was compressed or extended proportionally to the pressure, the cylinder being placed in communication with the interior of the working cylinder by a pipe, *B*, of sufficient size and fitted with a cock, *H*, by means of which the steam could be cut off from the instrument at any instant. So long as this cock was open, the indicator, if properly mounted, and the main steam piston were affected by precisely the same intensity of pressure, and the movement of the former was a measure of the pressure on the latter. A pencil, *s*, was attached to

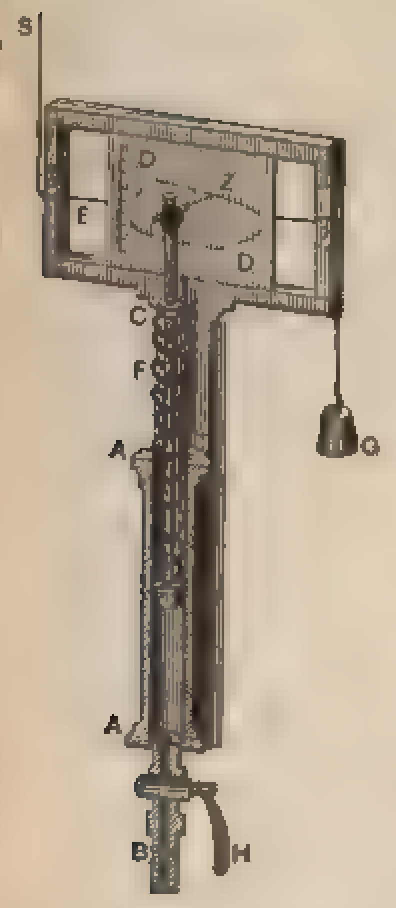

FIG. 12.—THE WATT INDICATOR

* See Tredgold on the Steam engine. London, 1827.

the indicator-piston, and its point recorded all such variations of steam-pressure on a movable slate, *D*, which was so connected, through *SE*, with the mechanism of the engine as to move in exact coincidence with the main piston, and precisely at right angles to the line of motion of the pencil. Thus, the abscisses of the curve produced were proportional to the motions of the piston, and the ordinates of the same points in the curve gave the simultaneous pressures.

In the later instruments of McNaught and Hopkinson, metal cylinders revolving on vertical axes were substituted for the sliding panel of Watt's arrangement, and a much more compact instrument was thus made.

McNaught's indicator, which was in general use until about 1860, when the first of the more modern forms, that of Richards, was introduced, had the form seen in the sketch, as described by Rankine, about the above date.*

AB is the barrel. Its lower end, *A*, contains a small cylinder, fitted with a piston, which cylinder, by means of the screwed nozzle at *A*, can be fixed in any convenient position on either end of the cylinder. The communication between the engine cylinder and the indicator cylinder is made by means of the cock, *K*.

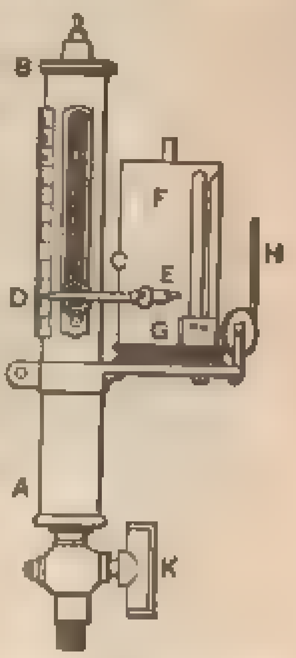

FIG 13.—MCNAUGHT'S INDICATOR

The upper end, *B*, of the cylindrical case contains a coiled spring, one end of which is attached to the piston, and the other to the top of the casing. The piston is pressed from below by steam, and from above by the atmosphere. When the pressure of the steam exceeds that of the amosphere, the piston is driven upward, and the spring compressed; when the pressure of the steam is less, the piston is driven downwards, and the spring extended.

A short arm *C*, a pointer *D*, which shows the pressure on a scale whose zero denotes the *pressure of the atmosphere*, and which is graduated upwards and downwards from that zero.

* Steam-engine, p 47 *et seq*

At the other side, the short arm has a longer arm, carrying a pencil *E*. *F* is a brass paper-drum, which rotates backward and forward about a vertical axis, and which, when used, carries a piece of paper called a "card." The cord *H* is to be connected with the engine in any manner which shall insure that the velocity of rotation of the drum shall bear a constant ratio to that of the engine piston.

The later devices have been introduced with a view to securing lightness of parts and reduced motion of piston.

Fig. 14 is a sketch, partly in section, of the first of the

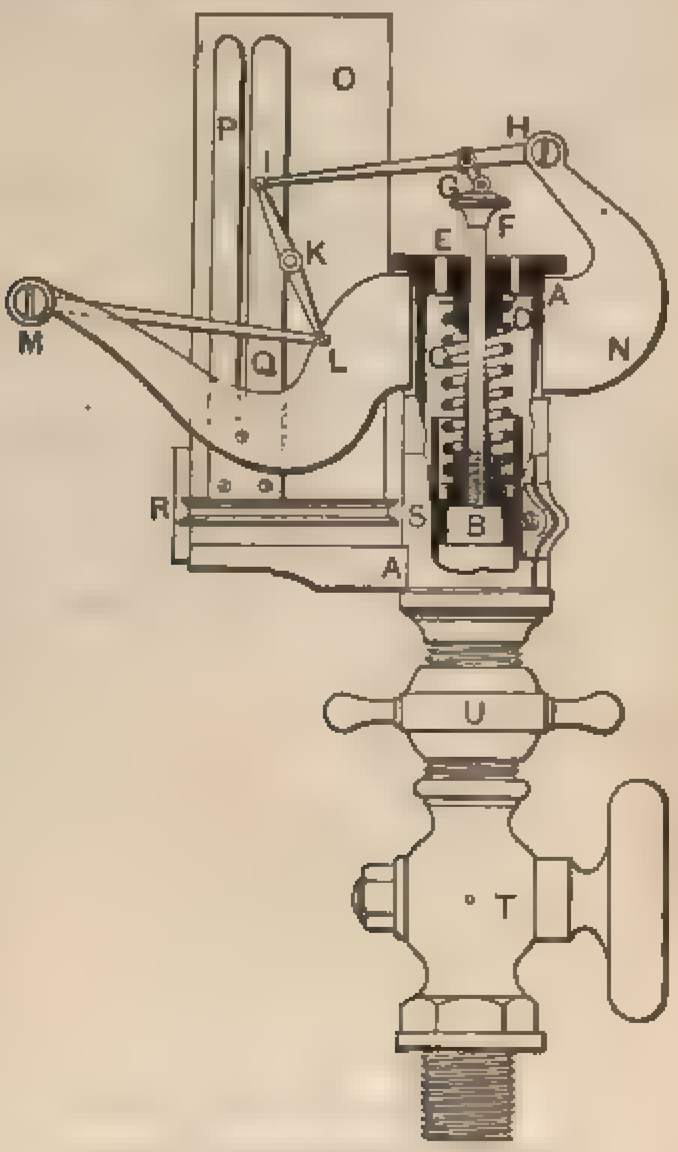

FIG. 14.—THE RICHARDS INDICATOR.

later type of instrument, the Richards indicator, invented by Professor C. B. Richards about 1860. *AA* is the cylinder, *B* is the piston, connected by a properly made spring, *CD*, with

the cap, *E*, of the barrel. The head of the piston-rod, *F*, is attached by a link, *G*, to the lever, *HI*, by means of which a comparatively large motion of the pencil, *K*, is obtained without much movement of the piston and its attached parts, and consequently with but little inertia-effect. A parallel motion of the Watt type, *HI*, *KLM*, guides the pencil-holder, *K*, in a right line parallel to this path of the piston of the indicator. The paper is wrapped about the cylinder, *O*, and secured at its ends by the clamps, *PQ*. The paper-cylinder is turned on its axis by a cord on the pulley, *RS*, which cord is attached to some form of "reducing motion" which causes it to move with the engine-piston.

Communication with the engine cylinder is established by a steam passage through the cock, *T*, and the instrument is secured in place by the clamp *U*. When in action, this cock is opened; the indicator piston rises and falls with the varying pressure in that end of the engine-cylinder, and the paper-barrel rotates backward and forward as the engine-piston moves. When all is ready, the instrument being heated up and working smoothly, the pencil is pushed lightly against the paper, and a diagram is drawn, representing all changes of pressure and volume of the working fluid during the period of contact. This modification of the indicator was found to give satisfactory results up to a comparatively high speed, and its limit of efficiency was determined by the degree to which the lightening of its parts could be safely carried.

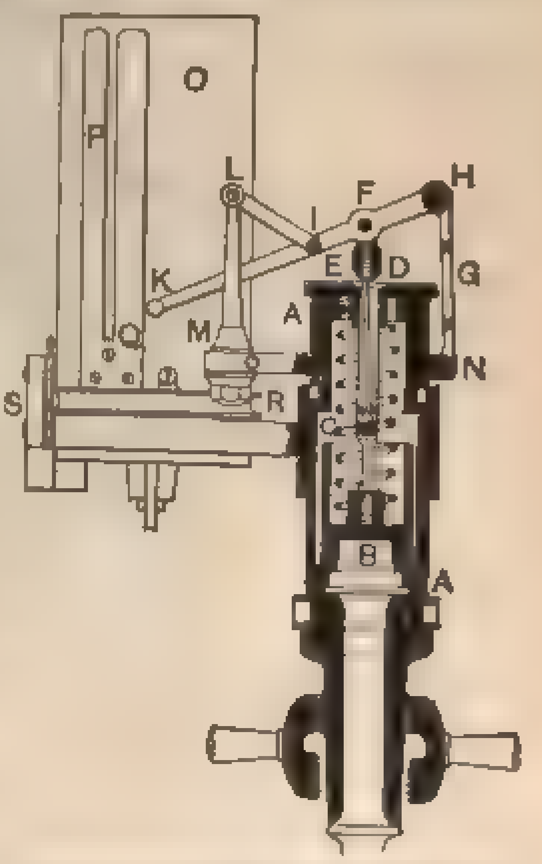

FIG. 15.—THE THOMPSON INDICATOR.

A still later form (1875) is that of Mr. J. W. Thompson, Fig. 15. In this indicator the same general style is retained, but the parallel motion is modified. The

cylinder, *AA*, contains a piston, *B*, connected by a spring, as before, to the cap, *DE*; while the head, *F*, of the rod actuates a pencil-arm, *HK*, and a parallel motion is obtained by linking on *LI* from the standard, *LM*, and *G* from the swivelling support *MN*, which also carries *L*. The action of the instrument when in use is precisely as before; its decreased weights of moving parts, however, enabled it to be confidently relied upon at speeds far above those of even the Richards instrument. The old McNaught indicator became unsatisfactory at about 60 revolutions per minute; the Richards carried this limit well up toward and sometimes above 200 revolutions; while the Thompson indicator was found capable of doing good work on even the fast engines of the most modern type at the date of its invention.

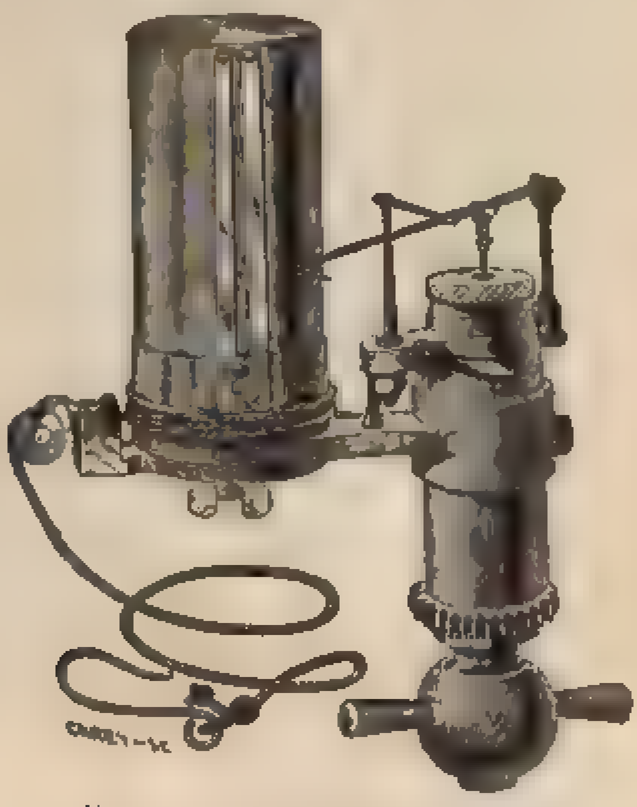

FIG. 16.—THE THOMPSON INDICATOR.

The most recent and a still lighter style of this instrument is shown in Fig. 16.

The later improvements consist in lightening the moving parts, substituting steel screws in place of taper pins, using a light steel link instead of a brass one, reducing the weight at the pencil-lever and elsewhere, shortening the length and reducing

the weight of the paper-cylinder one-half, and reducing friction to a minimum.

The paper-cylinder is so constructed that the tension of the coiled drum-spring within it can be varied for different speeds. As little or as much of the spring can be taken up or let out as desired, thus providing for fine adjustments.

Sufficient tension should be given to keep the cord taut at all points. When exceptionally accurate work is desired, the length of the diagram may be carefully measured, and compared with the length of a line traced on the paper when the engine is moved slowly. If the diagram is found to differ in length from this line, vary the tension of the spring till they agree.

All these indicators are provided with a piston 0.798 inch diameter, ½-inch area, and with springs for indicating pressures up to 250 pounds. When higher pressure is to be indicated, an extra piston 0.564 inch diameter, ¼-inch area, is used, which, when substituted for the other piston, doubles the capacity of each spring, thereby adapting the indicator for indicating pressures up to 500 lbs.

The Tabor Indicator, Fig. 17, the invention of Mr. H. Tabor

FIG. 17.—THE TABOR INDICATOR.

(1879), illustrates another ingenious attempt to evade all those difficulties incident to high speed of engines which have elim-

inated all the old forms of the indicator from the field. In this instrument the number of parts is still further reduced and the weight of such as remain is made as small as is thought safe. In the Tabor Indicator a stationary plate containing a curved slot is firmly secured in an upright position to the cover of the steam-cylinder. This slot serves as a guide and controls the motion of the pencil-bar. The side of the pencil-bar carries a roller which turns on a pin, and this is fitted so as to roll freely from end to end of the slot, with little lost motion. The curve of the slot is so adjusted and the pin attached to such a point that the end of the pencil-bar, which carries the pencil, moves up and down in a straight line, when the roller is moved from one end of the slot to the other. The curve of the slot just compensates the tendency of the pencil point to move in a circular arc, and a straight-line motion results. The outside of the curve is nearly a true circle with a radius of one inch.* The steam-cyclinder and the base of the paper-drum are made in one casting. Inside the steam-cyclinder is a movable lining cylinder, within which the piston of the indicator works. This cylinder is attached by means of a screw-thread at the bottom, and openings at the opposite sides at the top are provided for the introduction of a tool for screwing it in or out. Openings through the sides of the outer cylinder are provided to allow the steam which leaks by the piston to escape. The pencil mechanism is carried by the cover of the outside cyclinder. The cover proper is stationary, but a nicely fitted swivel-plate which extends over nearly the whole of the cover, is provided, and to this plate the direct attachment of the pencil mechanism is made. By means of the swivel-plate, the pencil mechanism may be turned so as to bring the pencil into contact with the paper-drum, as is done in the act of taking a diagram.

The pencil mechanism is attached to the swivel by means of the vertical plate containing the slot, which has been referred to, and a small standard placed on the opposite side of the swivel for connecting the back link. The connection between the piston and the pencil mechanism is made by means

* "The Tabor Indicator;" G. H Barrus, N Y, 1888

of a steel piston-rod. At the upper end where it passes through the cover, it is hollow and has an outside diameter measuring $\frac{3}{16}$ of an inch. At the lower end it is solid and its diameter is reduced. It connects with the piston through a ball-and-socket joint. A number of shallow grooves are cut upon the outside of the piston to serve as a so-called water-packing.

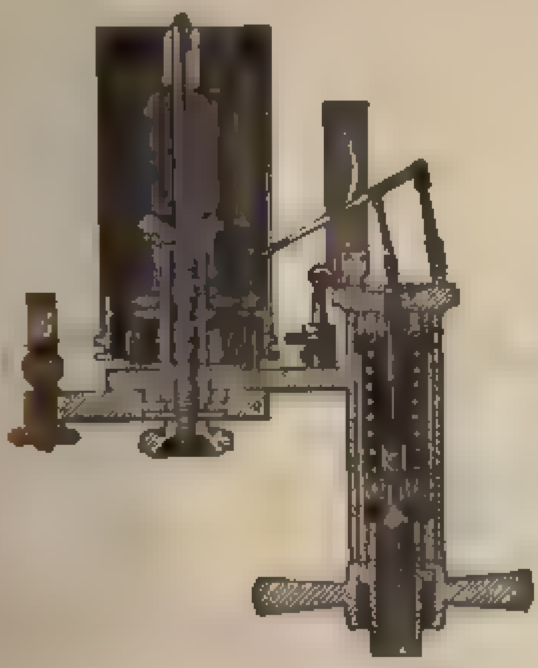

FIG. 18.—THE TABOR INDICATOR.

The springs used in the Tabor Indicator are of the duplex type, being made of two spiral coils of wire with fittings, as shown in the cut. The springs are so mounted that the points of connection of the two coils lie on opposite sides of the fitting.

The Crosby Indicator, Fig. 19, is still another successful recent type of the instrument (1879), and one which also illustrates that remarkable combination of lightness and accuracy which characterizes all good indicators. In this case, a still different form of parallel motion, light, stiff, and carefully adjusted, guides a very light pencil-holder carried at the end of a correspondingly light steel arm. The general arrangement of the indicator barrel and the paper-cylinder, with their attachments, is quite similar to those observed in the Richards and its successors.

If the conditions under which the spring acts be considered,

it is readily seen that, when the cord has the maximum other resistances to overcome, the drum-spring should offer minimum resistance. At the beginning of the stroke, when the spring is overcoming the inertia and friction of the drum, its resistance should be a maximum, and should gradually decrease. Here, a short spiral drum-spring is adopted, giving at the beginning of the stroke a comparatively slight resistance, which gradually

FIG. 19.—THE CROSBY INDICATOR.

increases until it reaches the maximum at the end of the stroke. In the other direction the recoil is strongest at the beginning of the stroke, and decreases to the end.

Duprez, Hirn, and Webb employ a screw, by means of which the steam is prevented from lifting the piston until the pressure exceeds a certain amount. Until this instant the indicator-diagram is a horizontal line; it then becomes curved. When

the screw is turned, the piston is again prevented from moving until the pressure exceeds a certain other limit, so that a series of corners is obtained, which are points on the real indicator diagram, and may be joined by hand. When vibrations of the spring are in this way destroyed, exactly the same indication may often be obtained during four or five successive strokes.*

Mon. Hirn would prefer, where practicable, a directly connected spring, of considerable amplitude of range, stretching and compressing it by means of the screw just described, and allowing the attainment of the pressure registered at any instant to be indicated by the slight vibration or jump permitted by the lost motion in the grip on the spring as the steam-pressure passes that point. In these indicators it is evident that the

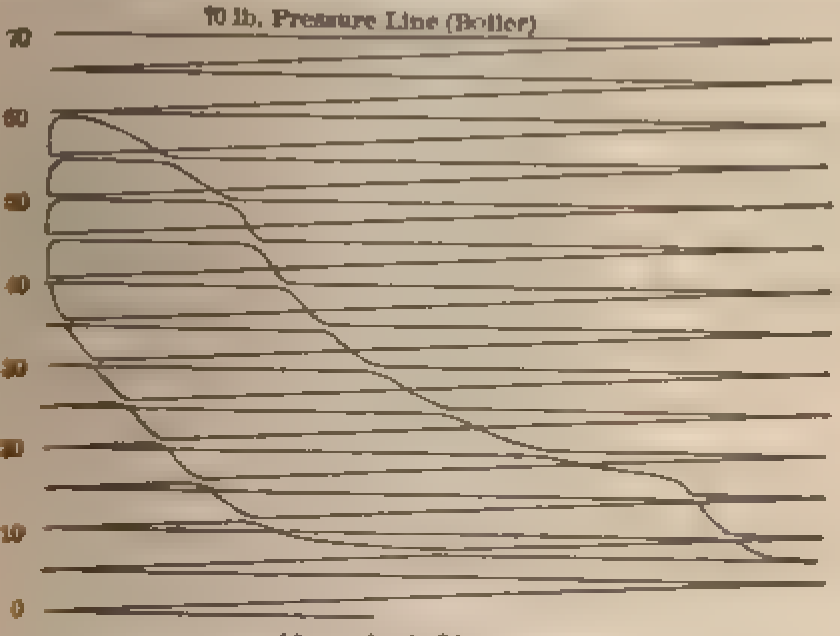

FIG. 20.—WEBB'S DIAGRAM.

diagram produced is then a "composite" of a number of successive indications, taken in as many successive revolutions of the engine. At the high speeds for which only such instruments are designed this is probably no disadvantage; but the instrument cannot show what occurs throughout any one revolution. The screw is usually so attached, however, that it may be readily removed at any moment, and the indicator thus quickly converted into the common form.

* Perry, The Steam Engine, 1874. Bulletin de la Soc. Ind. de Mulhouse, 1876. London Engineering, Dec. 14 1888, p. 576.

The indicator of Professor Webb is intended for use on fast-running engines where the inertia of the parts of the standard type of instruments is embarrassing.

Fig. 20 is a diagram taken at 400 revolutions per minute. The series of zigzag lines is the blank diagram. Each line is made by one stroke of the engine, and at 400 revolutions it would be about three seconds before the diagram is finished. If the indicator cock is open, the pencil will make the diagram. Instead, at the start, of following the 55-lb. line, it jumps up until the pressure on the cylinder falls again to 55 lbs., when it will come back to the line and finish it. The pencil will then return to the left side on the diagonal line, and the process will be repeated until the card is complete.

In Fig. 21 an indicator is shown with this device attached.* The frame of the instrument is extended up to *e*, and a hole is made through it in which the screw *b* slides freely. This screw has a nut, *e*, which is confined between the forks of the frame. The lower end of the screw has a forked head, *c*, which embraces the upper end of the piston-rod, and is attached to the same by means of a pin, which passes through holes in both; the hole in the fork is, however, slotted out a little larger than the pin, as will be seen hereafter. If the nut, *e*, is revolved, the screw will draw up the piston against the spring contained in the indicator cylinder until we have set the latter at any desired pressure. If this pressure be above the maximum pressure in the engine cylinder, the pin at *c* will remain resting on the bottom of the slot; but if it be below, then during any stroke of the engine it will remain there only so long as the cylinder pressure is below that of the spring; it will jump to the top of the slot when the former exceeds the latter. The slot is made the right length to allow of a jump of 4 or 5 lbs.,

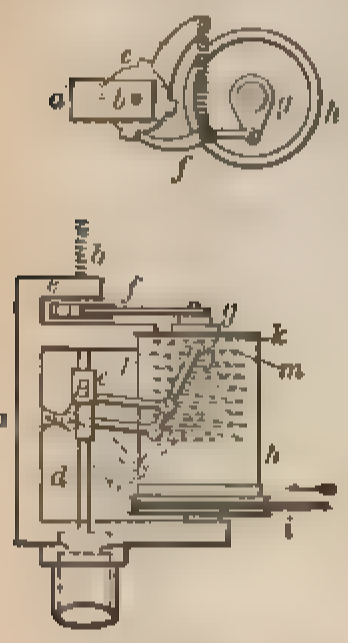

FIG. 21.—THE WEBB INDICATOR.

* Trans. Am. Soc. M. E., vol. 11

as shown. To complete the arrangement we add the pawl f and crank g mounted on the top of the indicator-drum, h, and so arranged that during each backward stroke f shall revolve c, and thus let the screw, b, with the piston and spring, gradually down.

This instrument thus produces the mean of several diagrams, making the whole by combining parts of each.

The indicator is sometimes given the form shown in the accompanying engraving, in order to obtain a record of the

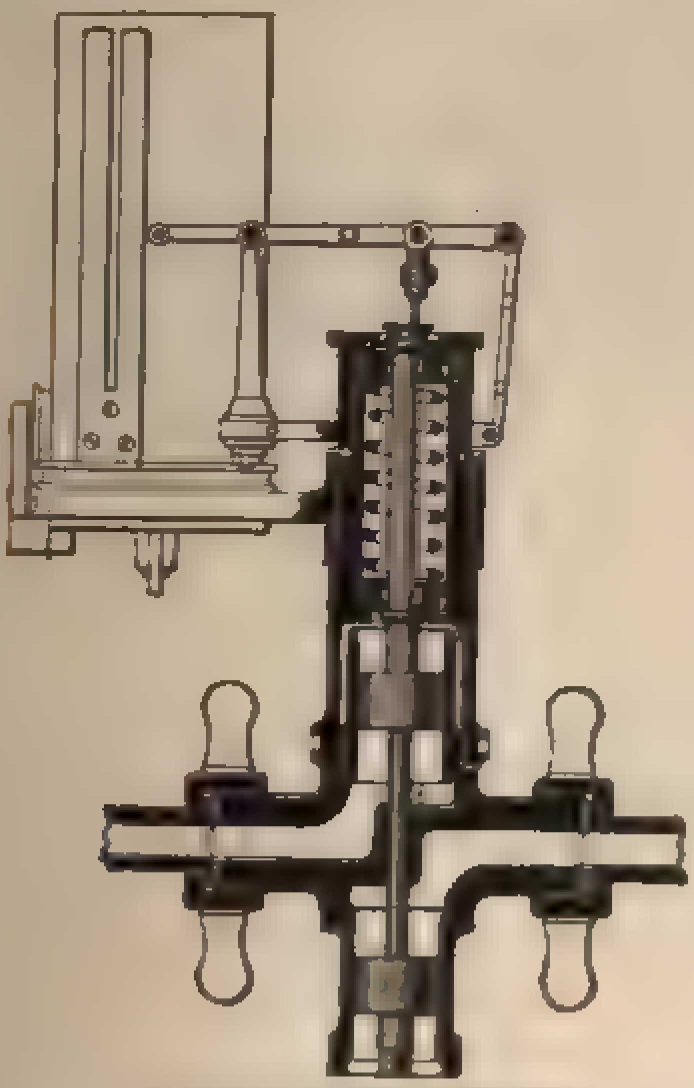

FIG. 22.—THE DOUBLE INDICATOR.

net pressure on the piston. One piston receives the steam-pressure at one end of the steam-cylinder: the other is acted upon by the steam on the other side of the piston. The effort on the spring is thus the net pressure, and the diagram produced, as shown in the next figure, presents this net pressure

at every instant throughout the stroke. The upper portion, *abkdl*, and the lower part, *fgedk*, measure the work done in the forward and backward strokes, respectively.

Every modern indicator thus combines the essential elements of construction which have been seen to be required to secure accuracy of diagram, and in a very admirable manner. The latest and best may be relied upon to give good "cards"

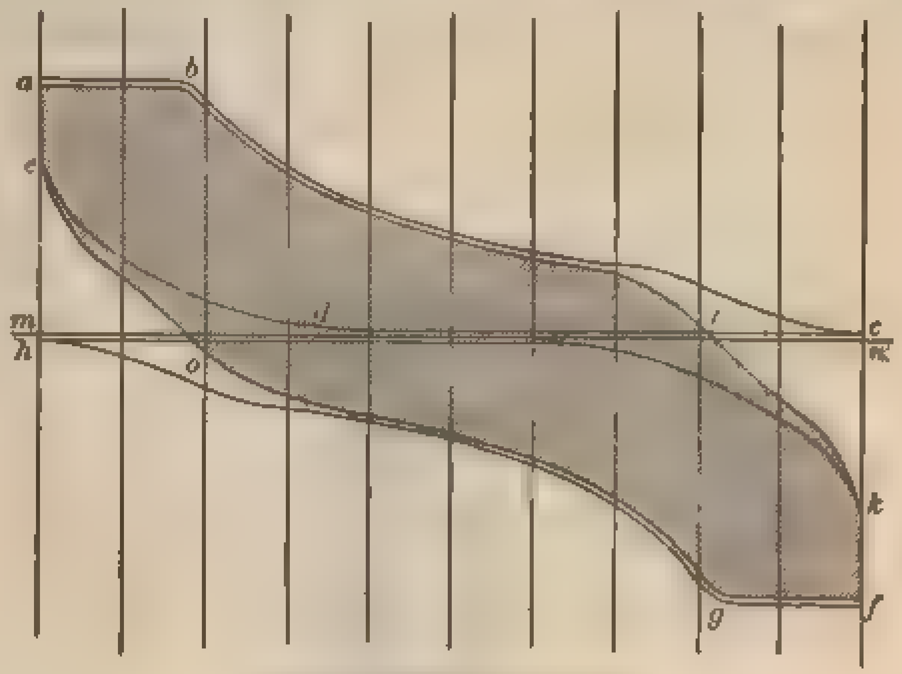

FIG. [illegible].—DOUBLE DIAGRAM.

at any speeds yet attained in usual operation by the fastest of the "high-speed" engines.

54. The Standardization and Test of the Indicator should always precede its use. To give satisfactory results, the instrument must give a diagram of which the abscissas shall exactly represent the successive positions of the moving piston, while the ordinate of each as exactly measures the simultaneously occurring pressure within the working cylinder. The weight and inertia of every moving part of the indicator introduce errors which, while they may not be completely eliminated, may, by reduction of size and special expedients, at least be rendered so small as to be unimportant at any ordinary speeds. They are, however, more difficult of prevention as the speed of rotation and the pressures adopted increase. In a well-made instrument the spring will precisely measure pressures, the

pressure in the indicator will be sensibly the same as in the engine, and its piston will move so freely as not to affect the indications by its resistances due to close fitting. In all, however well made, on the other hand, it is found impossible, by any art of design or construction, to wholly eliminate the influences of inertia of moving parts, friction of joints and guides if the latter be used, and of piston or pencil, or the effects of variation of spring tension on the motion of paper cylinder and card. Standardization is the process of detecting and measuring such errors as are observable in the action of the apparatus, and the determination of their influence on the indications obtained by its use.

Springs are tested most satisfactorily by connecting the indicator, with its appropriate spring in place, with a small steam-boiler or steam-reservoir or convenient steam-pipe, in such manner that simultaneous measurements of pressure may be taken by the indicator record and a standard test gauge known to be correct. If the spring be tested cold, it will be found inaccurate if it had been found right when hot; and the correct reading of a cold spring is evidence that it is not right under steam;* since, when the indicator is in use and the spring heated, both by the steam leaking past the piston and by conduction through its attachments from the piston and from the indicator-barrel, its strength and its elasticity are sensibly modified, the spring being thus weakened. In thus testing springs, such arrangements should be made as will enable the observer to hold the pressure at any desired point until readings from the standard test-gauge can be deliberately and precisely taken. If the spring is found unreliable, it should be at once exchanged for a good one.

"Throttling," or loss of pressure between the engine and the indicator, is produced by long, tortuous, or contracted passages. The connections should be as large, as straight, and as short as practicable, and, other things being equal, that indicator is best in which steam connections can be best effected.

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* This difference has been found to amount to 2½ or 3 per cent.—Proceedings of Brit. Inst. C. E., 1885 Brightmore, discussion.

The cock under the indicator should have an opening fully as large as the pipe itself. It should also have a hole bored in from one side for the purpose of freeing the instrument and connections from water of condensation coming over with the steam. The holes in the cock, or the upper part of the barrel of the indicator, should be large enough to permit free egress to any steam that may leak past the piston; and the latter is, in all good makes, so loose as to leak observably under pressure. The effect of leakage is insensible; but were the fit a tight one, the resulting friction might be important. It is to insure this exact correspondence of pressure in the working cylinder of the engine and in that of the indicator that it is customary, on all high-speed engines, to employ indicators simultaneously at both ends, thus obtaining very short and direct steam connections.

The springs should be so made and fitted that their action under pressure, and when in use, may not throw the piston out of line or cramp it in the barrel, and thus produce what are sometimes found to be serious errors. A small leak does no harm, and is, on the whole, desirable. The piston should move so freely that, the spring being removed, the breath may blow it from end to end of its barrel and draw it back again.

The friction of the pencil on the paper is probably closely proportional to the force with which it is pressed upon the latter, and variable with the texture of the paper and the sharpness of the pencil-point, and its material. This is often an important source of error in the diagram. The reacting effect of this pressure on the pencil mechanism is also, but in comparatively slight degree, a source of inaccuracy of record. The result of such frictions is the production of an enlargement of the "card" to the extent, often, of a very appreciable, and sometimes of an important, amount. This friction is sometimes relied upon to diminish those oscillations of the instrument which at high speeds render the diagram difficult of measurement, or even untranslatable. In such cases the real power of the engine may be several per cent. less than shown by the in-

strument. To avoid this difficulty, it is best to make the pencil bear on the paper only just hard enough to make a visible mark. A hard-lead pencil or one of soft metal smoothly pointed, paper having a "metallic" or glazed surface, and a light, steady pressure producing an extremely fine but perfectly visible line are the conditions to be sought. In such case, the error due to friction of pencil will be inappreciable.

The stretching of the cord turning the paper-barrel is a common cause of inaccuracy in length and of distortion of the diagram. The varying tension of the spring, and the surges due to the inertia of the rotating mass, together cause variations of length of the cord that may give rise to errors of really important magnitude. The string, even when its primitive stretch is taken out of it by a preliminary application of a heavy load, retains some elasticity, and will have a sensibly variable length under the constantly varying pull when in use. Any observable friction of the paper-barrel also tends to exaggerate this action. The inertia of the drum tends to compensate this effect, and it is possible to so adjust the strength of the spring to this inertia as to make the variation of stress on the cord comparatively small.

The effect of this stretch is to cut off a part of the diagram at one end, and it is perfectly possible thus to reduce the apparent indicated power of the engine 10 or even 20 per cent. below the correct quantity.* The longer the cord, the greater the error; and differences of sensible amount may often be detected between the diagrams from opposite ends of the same cylinder, in area of diagram and in point of cut-off and amount of expansion, produced by differences in length of the cord used. The higher the speed of rotation of the engine, the greater the amount of this error; and instruments giving perfectly satisfactory cards at low speed may produce very defective diagrams at high speed. The lighter the drum and the spring found practicable, the better the results. A cord should always be well stretched before use; but a fine steel wire

* Proc. Brit. Inst. C. E., 1885: Brightmore.

is much to be preferred.* The paper-cylinder or drum should have carefully adjusted springs for fast work. The difference in its initial and final tension should be as nearly as possible equal to the inertia-stresses of the drum. Improvement has been carried so far in the reduction of weight of paper-barrel as to bring it, in one case at least, as low as 209 grammes (3088 grains). It has been often proposed to use aluminium for moving parts in order to reduce inertia-effects to a mininum. Errors due to this action, in good instruments, fall much below 1 per cent. In adjusting the instrument, the tension of the drum-spring should vary as the square of the number of revolutions; and it should be set for any speed in such manner that the length of the diagram should be the same at starting as when at full speed, as nearly as possible.

FIG. 83.—DRUM-TESTING APPARATUS.

In testing the action of the drum-spring, either of two or three devices now in use may be employed. The accompanying sketch shows that devised by Mr. Brown.

This instrument was designed to show the strains on the cord. From its diagram may be calculated the errors due to the stretch of the cord. The testing instrument consists of a plate, *A*, to one end of which is fastened the frame, *BB*, carrying the slide, *C*, and its cross-head, *D*. The spring, *F*, is screwed to the cross-head; the other end is connected with the lever, *G*, carrying the pencil. The rod, *E*, which moves the slide, *C*, receives its motion from a crank not shown. The swinging leaf, *F*, holds the paper on which the diagram is to be taken. The indicator is clamped to the plate, and the drum-cord connected with the spring. The crank is

* Mr. Wallace finds the yield of a good ordinary stretched indicator-cord to be from 0.008 to 0.0125 per foot per pound, and of wire No. 36 B. W. G. 0.003.

made to move at the speed desired. The paper is then raised to the pencil, and the diagram taken. If the strain on the cord is constant, the forward and return strokes will be parallel: but if the strain is not constant, the pencil will rise and fall as the strain varies. The line below the diagram is the line of no stress, drawn when the cord has been detached from the indicator.

The diagrams are shown two thirds their original size.

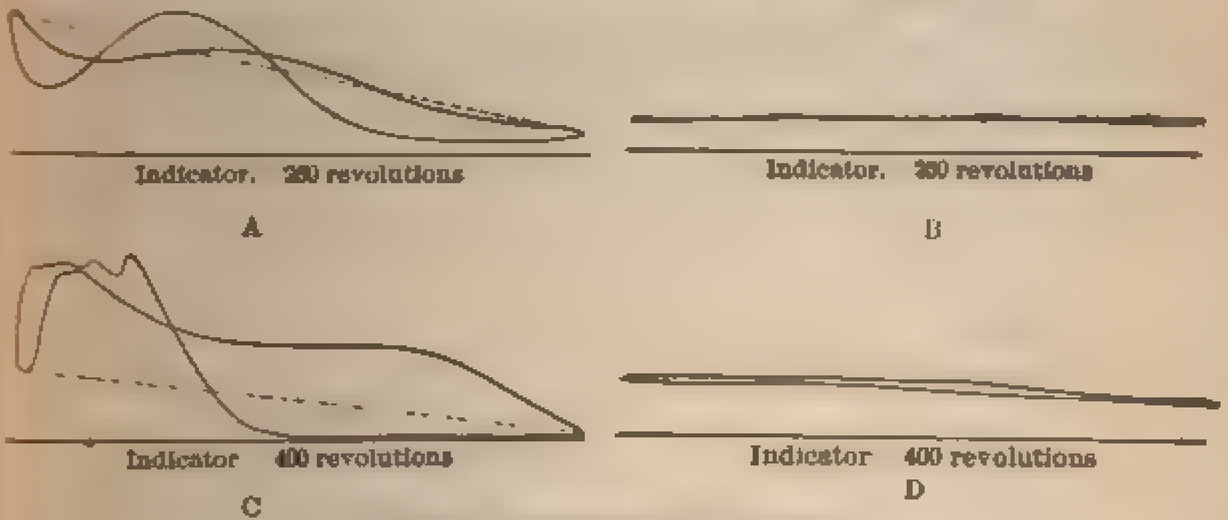


FIG. 24.—DRUM STRESS DIAGRAMS.

Oscillations of the pencil about its proper position, and the consequent production of wavy lines in the diagram, cause the most serious defects in diagrams taken at high speeds of engine. Such deformations of the diagram are due to the inertia of the pencil and its holder and connections, and become greater as speeds increase, until, with every instrument, a speed is finally reached at which the diagram becomes unintelligible, as in the figure on page 148, which represents the card obtained by using an old style of indicator, with a light spring, at 300 revolutions per minute. With the best modern indicators it is easy to secure a perfectly smooth diagram at this speed.

These vibrations are the more serious as the proportion of the weight of the moving parts of the indicator are the heavier and as the spring is lighter. Their effect is not only to disguise the true form of the diagram, but also to enlarge it and thus to give too great values of power developed. Professor Reynolds gives the following as speeds at which this variation becomes one per cent. in an indicator having a piston-area of one-half

square inch and a weight of moving parts equivalent to 0.33 pound at the piston:*

Spring used, lbs. per sq. in.	No. revolutions per minute.
20	166
40	237
60	288
80	332
100	371

This error varies directly as the weight of the moving parts. In modern indicators of the best forms, it is probably inap-

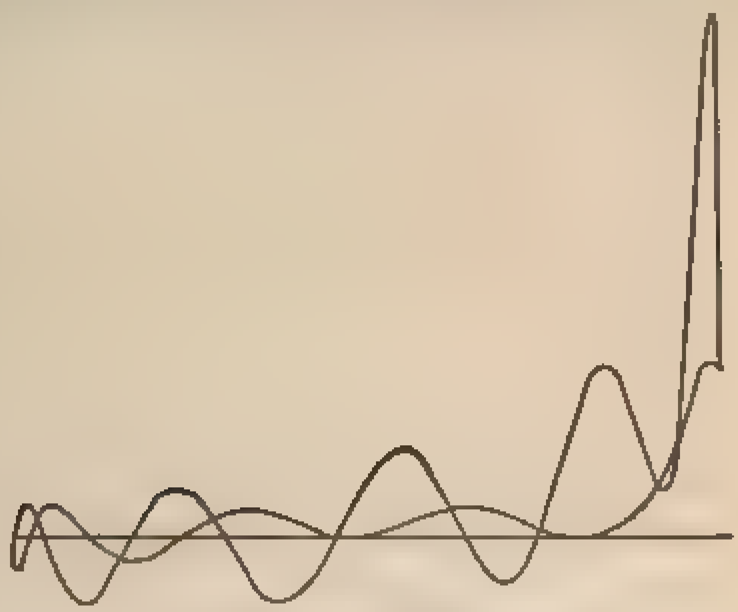


FIG. 25.—INDICATOR VIBRATIONS.

preciable at all familiar engine speeds, the maximum being taken at about 300 revolutions per minute, five per second; and it may be assumed by the engineer that if his indicator is of good make, if he finds its parts correctly made, and if he keeps them in good order, he may rely, in all ordinary cases, on obtaining diagrams correct to within the limits of his nicest measurement, if the instrument is properly attached to the engine and skilfully handled.†

For higher speeds than are now obtained, indicators of the class illustrated by that of Duprez must be employed.

* Proceedings Inst. C. E., vol. lxxxiii. 1885.

† Barrus on Modern Indicators, and Discussion thereon: Trans. Am. Soc. M. E., vol. v. pp. 310–339. 1884.

The following is a description of Professor Reynolds' device for checking the movement of the drum, and to ascertain what distortion is caused by its irregular fling and friction: *

A Grove battery of five cells, in conjunction with a Ruhmkorff coil, is used. The wire from one pole was connected with one of the binding-screws (*H*) of the coil as usual, but the wire from the other pole of the battery was connected with the engine. A wire from the other binding-screw (*G*) was attached to the contact-breaker (*B*), a smooth piece of wood, into which pieces of wire were inserted at equal distances, the distance between the first and last wire being the length of the stroke of the engine. This was fixed on the lower slide, so that a pointer (*A*), secured to the cross-head, should slide on it. One

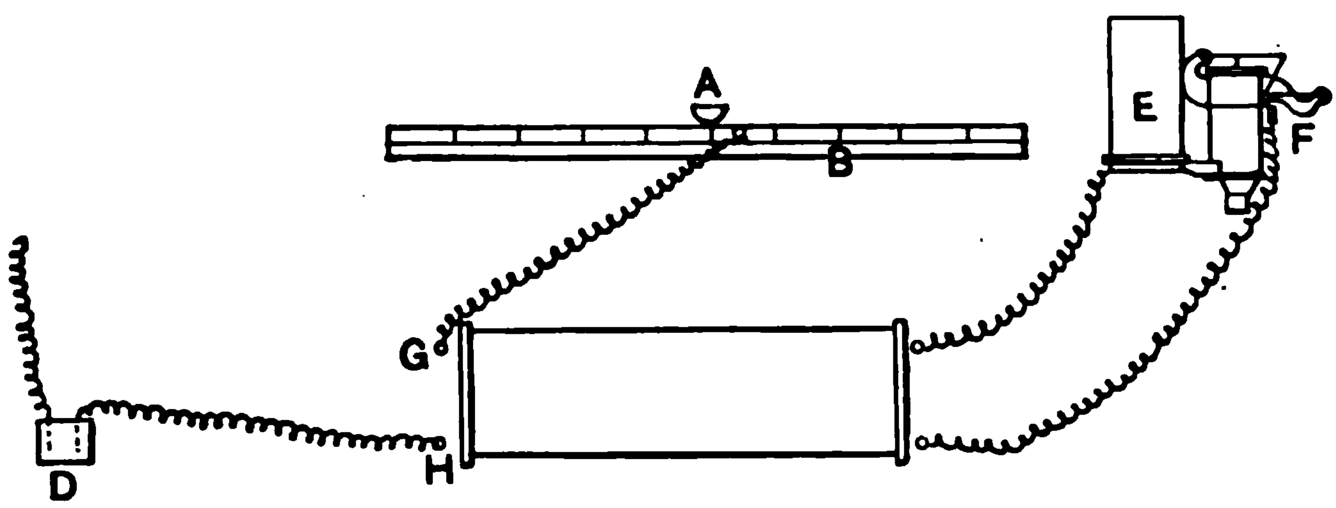


FIG. 26.—DISTORTION BY STRETCH OF INDICATOR-CORD.

wire of the secondary coil was connected with the drum (*E*), and the other to a cup of mercury, into which the metallic pencil (*F*) dipped, thus completing the circuit when the pencil touched the paper.

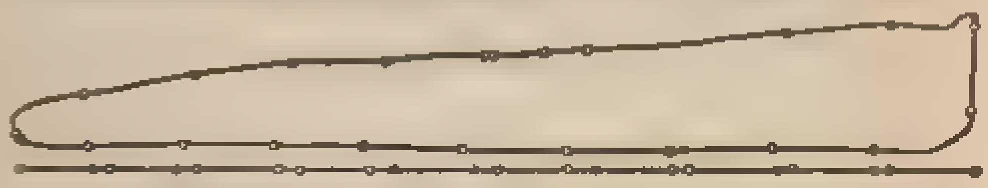
In the following diagrams the relative positions of the circles show which parts of the diagrams are lengthened, and which are shortened. The effect is not merely to shorten the ends and lengthen the middle of the diagrams, but also to distort them, *i.e.*, to cause corresponding points not to lie in the same vertical line. The amount of this distortion is shown by the distance between corresponding points on the atmospheric line. (Fig. 27.)

It is evident that the pencil may be made barely to touch

* Proc. Brit. Inst. C. E, 1885, No. 2070.

the paper without marking it and the diagram made by the sparks alone.

Front-end pricked diagram taken with wire at 107 revolutions.



Front-end pricked diagram taken with string at 107 revolutions.

Front-end pricked diagram taken with wire at 127 revolutions.

Front-end pricked diagram taken with string at 127 revolutions.

FIG. 27.—ELECTRIC DIAGRAMS.

Comparisons of indicators will often eliminate uncertainty as to their reliability. If an indicator is known to be right, the diagrams produced by it should be, under similar conditions, duplicated by an instrument the accuracy of which is doubted. Where three or more instruments are compared, the presumption is usually a fair one that, if one differs in any important degree where the others agree, it is defective. Where several are to be compared, it is sometimes practicable to take diagrams from them all simultaneously by fitting up properly. In such cases, all should be equidistant from the steam-cylinder and should have equally straight and large connecting pipes. One large pipe and cock, taking steam from the cylinder, terminated by the several pipes leading radially from its top to the several instruments, will usually answer the purpose.

In comparing the details of construction, it is to be remembered that the points to be studied are the exactness and per-

fection of dimensions and workmanship, and the weight of parts and their action as affected by inertia. The latter point has been seen to be peculiarly important when the instrument is to be used on engines at speeds exceeding about a hundred revolutions per minute.

Comparing, in this particular, the action of the paper-cylinders, or -drums, we find the time of a vibration, the forces freely acting, to be

$$t = 2\pi\sqrt{\frac{d}{a}};$$

in which d is the angular displacement, here to be reckoned from the position of mid-throw, and a the angular velocity. Then

$$a = \frac{4\pi^2 d}{t^2};$$

and

$$d = \frac{1}{2}\,\frac{\text{length diagram}}{\text{radius of drum}} = \frac{l}{r},$$

and the couple acting to start the drum into harmonic motion is

$$aI = \frac{4\pi^2}{t^2}\, I \times l.$$

That instrument, therefore, which has the least value of I the moment of inertia of the drum, is least liable to inaccuracy from this source of stress. The best adjustment should be sought in each case, and the comparison effected after this adjustment has been made. Since the effort of the drum-spring is usually directly proportional to the angle of motion, and since the force due acceleration is zero at mid-throw, if the difference of tension on the cord at the beginning and end of the motion is twice the effort required to overcome the inertia-resistance of the drum at starting, the action on the cord will be uniform when at speed, and the diagram entirely free from distortion from this cause. Other things being equal, that is

the best instrument in which this adjustment is secured. In some cases they are arranged for such adjustment at several usual speeds, or so as to permit the tension of the spring to be altered as desired—increasing it at high speeds, diminishing it at lower speeds. The acceleration for a $4\frac{1}{2}$-inch card is $\frac{0.11523}{t^2}$, or for a 3-inch card $\frac{0.7703}{t^2}$; and, for the power, a pound acting at the circumference of a drum 2 inches in diameter will give the following maximum resistance at the stated speeds:*

Revs. per min.	R.	Revs. per min.	R.
120	0.5	360	4.2
180	1.0	480	7.4
240	1.8	600	11.5
300	2.9	1000	32.0

These stresses evidently become serious at high velocities, increasing, as they do, as the square of the speed; and the higher the speed the greater the difference in favor of that instrument having lightest drum.

The stretch of the cord used (if sensible) should be observed; as this is an element which determines also the amount of distortion of the diagram due to the inertia of the drum and the action of its spring. In all cases we have the moment of the pull, P, on the cord,

$$Pr = \frac{d^2\alpha}{dt^2} + Rr + Fr;$$

in which the angular acceleration is $\frac{d^2\alpha}{dt^2}$, and Rr and Fr are the spring and the friction moments. The first quantity has been seen to be zero at mid-throw, its value increasing each way: the friction moment may be taken constant and unimportant, and the spring-resistance variable with its flexure, as already seen. The stronger and the less elastic the cord, and the better the adjustment of the spring-action to the inertia-effect, the

* Wallace on the Indicator: Trans. Inst. Scotland, 1888, p. 3.

more accurate the diagram. In good examples these effects are unimportant.

Comparing the indicators as to the effect of surges and oscillations, it is found that both these actions may become serious at high speed and with heavy pencils, springs, and pistons. The surge of the moving parts due to their rise or fall through the height of the diagram tends to increase the area of the curve. If indicators give similar and correct results at moderate speeds, this increase at higher speeds may be compared to determine their relative merits. It should never equal 1 per cent. This limit is found by Professor Reynolds, for the Richards indicator, at the speeds already given. It is seen that the maximum speeds of rotation in revolutions per minute is about

$$R = 40 \sqrt{s},$$

where s is the scale of the spring in pounds to the square inch; and this disturbance varies directly as the weights. This comparison may therefore be effected by weighing the moving parts.

The vibratory disturbance of the pencil is due to the elasticity of the spring, and its time is

$$t = 2\pi \sqrt{\frac{wr^2}{12pmg}};$$

where wr^2 is the effect of the moving parts reduced to the working-point; p, m, and g are the total load on the piston, the ratio of pencil and piston motion, and the acceleration of gravity. These disturbances are not serious as affecting the area of the diagram, but they are sometimes important as obscuring its meaning. A comparison of indicators in this regard would be made by taking diagrams at continually increasing speeds and noting the point at which the outlines of the figure become wavy and when they interfere with its legibility. It is seen that this defect increases as the square root of weight of moving parts, and is the more serious as the weights are nearer the

pencil and subject to rapid movements and quick changes of direction of motion. These irregularities may be partially controlled by the pressure and friction of the pencil, but only at the sacrifice of accuracy. The value of t should always be as small as practicable. If too large, the diagram may be seriously distorted. Any number of oscillations in the tracing of the card exceeding 25 or 30 may be permitted; less than 20 or 25 is objectionable.

Comparing springs, it will be often found that considerable differences are observable in their indications, both cold and hot. They should be examined to see that they take no permanent set, that they yield in exact proportion to the pressure, and that their attachment to the instrument is such as not to produce lateral strain or friction. Springs which have been already repeatedly given their full set by the makers are best. They should always be tested and compared hot.*

The best indicator, as is now evident, is that which, by such comparison and examination as has been described, is found to give the most exact and reliable diagram, and to be least affected by inertia-forces and the action of its own parts at high speed; it is that, in detail, which has proportionally the largest and lightest piston, the stiffest and lightest springs, the least friction of moving parts, the most perfect pencil mechanism, the most accurate and constant scale of pressures, the most perfect adjustment of drum spring, and the lightest moving parts generally.

The following is the method of comparing indicators adopted by the Navy Department: †

A horizontal pipe, 2 inches diameter and 24 inches long, fitted with suitable pipes and valves for the admission and discharge of steam and provided with three nipples, two for the attachment of indicator and one for a steam-gauge, was used in the tests.

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* See the valuable papers of Dr. Berndt on this subject in the Sächsische Ingenieur und Architecten Verein, 1882–85 meetings · and Lond. Engineering, 1877–8

† Report of Chief of Bureau of Steam Engineering, 1888.

The steam-gauge having been secured in place, steam was blown through the test-pipe and indicator-nipples several times to free the pipe of water and dirt.

The indicators, after being well oiled, were secured in position and steam admitted, the pressure being allowed to rise until the limit for which the springs were designed was reached, in order to bring the instruments to their working temperature.

After trying the instruments to see that their movements were free, steam was discharged from the test-pipe and the indicator-cocks closed. The piston of each instrument was then pressed down slightly by hand and allowed to return to its normal position, with the friction of the moving parts opposed to the movement of the spring. When this had been done, the atmospheric line was drawn across the card. Steam was then admitted to the instruments and so regulated that the hand of the steam-gauge would rise slowly to the interval of pressure to be noted: and when it reached that point, at the word "mark," an operator stationed at each instrument drew the required line of its scale. All lines of the scales were drawn in the same manner, the top steam-line of the first test of each series being extended across the card.

Before beginning the down scales, the steam was allowed to rise a pound or two above the pressure to be first noted, in order to oppose the friction of the instrument to the movement of the spring. At the end of the down scale, the steam was shut off and discharged from the pipe and the indicator-cock closed before drawing the atmospheric line.

To determine the comparative indications of identically the same power by the two instruments, the following method was used:

The indicator-pipe at the outer end of the engine was fitted with a T and two right-angle branch pipes of equal diameters and lengths terminating in nipples. To the latter, the indicators were attached, after clearing the pipes of water and dirt and lubricating the cylinders. The springs of the paper-drums were adjusted to approximately the same tension. The cords

around these drums were tied to each other and to a single cord connecting with the indicator motion. This arrangement gave coincident motion to both drums without sensibly affecting the lead of the cords, as the angle between the latter was small.

One operator could readily take cards from both indicators at the same time.

Ten cards having been taken from each indicator, the latter were interchanged and then ten more cards taken from each. This change was made in order to eliminate any errors due to possible differences in the bore and lead of the branch pipes.

The test to determine the pencil movement was made as follows:

The spring of each indicator having been removed, a micrometer gauge was fitted to the cylinder and the weight of the piston and attachments taken on the end of the micrometer screw, the zero of the wheel coinciding with that of the vernier. A line was then drawn with the pencil of the instrument and formed the first one of the scale. The micrometer screw was then turned one revolution and a second line drawn, this was repeated until the scale was complete for the movement of the piston.

A test to determine the line of motion of the pencil in each instrument was made as follows:

The spring having been removed, the piston was pushed up its entire stroke, the pencil at the same time drawing a line on the card, while the paper-drum was securely held by the detent attachment. Ten such lines were drawn on the card with each instrument.

All moving parts of both instruments were carefully weighed and their weights in Troy grains found.

At the conclusion of the foregoing tests, the steam-gauges used in the work were carefully compared with the mercury column.

The test of indicators, taking simultaneous diagrams from the same end of the engine, is liable to give misleading results unless great care is taken to have both equally well fitted and

similarly situated. To insure perfect fairness, they should be transposed and again compared. The following is the result of a comparison so made. The results can of course only be taken as gauging the work of the individual indicators so compared:

TESTS OF INDICATORS.

Simultaneous cards taken from engine with A and B indicators fitted with 20-pound springs.

Number of Cards	Mean pressure. A	Mean pressure. B	Difference	Indicated horse-power. A	Indicated horse-power. B	Difference
1	13 30	12 65	65	26 797	25 487	1 310
2	13 80	13 20	60	27 803	26 595	1 208
3	13 65	12 85	80	27 502	25 890	1 612
4	13 45	12 65	80	27 099	25 487	1 612
5	13 30	12 55	75	26 797	25 286	1 511
6	13 47	12 60	87	26 942	25 203	1 739
7	13 62	12 58	1 04	27 243	25 163	2 080
8	14 20	13 25	95	28 403	26 503	1 900
9	13 90	12 85	1 05	27 803	25 702	2 101
10	14 30	13 30	1 00	28 603	26 603	2 000
1	16 95	16 16	79	33 161	31 615	1 546
2	15 80	14 60	1 20	31 372	28 990	2 382
3	15 50	14 28	1 22	30 664	28 250	2 414
4	15 45	14 10	1 35	30 677	27 997	2 680
5	17 00	15 65	1 35	34 003	31 303	2 700
6	15 25	13 90	1 35	30 503	27 803	2 700
7	14 20	13 00	1 20	28 403	26 003	2 400
8	16 20	14 70	1 50	32 403	29 403	3 000
9	15 20	13 80	1 40	30 403	27 603	2 800
10	12 65	12 35	30	25 303	24 702	601
Means	14 5595	13 5510	1 0085	29 0942	27 0794	2 0148

Power measured by A 7 4403 per cent. greater than measured by B.

Simultaneous cards taken from engine with A and B indicators fitted with 40-pound springs.

Number of Cards	Mean pressure. A	Mean pressure. B	Difference	Indicated horse-power. A	Indicated horse-power. B	Difference
1	14 60	14 30	30	29 842	29 229	613
2	14 80	14 00	80	30 251	28 616	1 635
3	13 90	13 20	70	28 412	26 981	1 431
4	13 50	12 75	75	27 594	26 061	1 533
5	14 60	14 35	25	29 643	29 122	521
6	13 00	12 45	55	26 572	25 448	1 124
7	12 15	11 85	30	24 835	24 221	614
8	12 52	12 50	02	25 591	25 550	041
9	12 70	12 20	50	26 144	25 115	1 029
10	12 50	12 00	50	25 550	24 528	1 022
1	15 70	15 10	60	33 008	31 746	1 262
2	14 00	12 75	1 25	29 025	26 433	2 592
3	14 35	14 00	35	28 886	28 207	679
4	17 00	16 43	57	33 259	32 144	1 115
5	14 75	14 60	15	29 716	29 426	300
6	18 40	17 25	1 15	36 515	34 252	2 283
7	17 20	16 50	70	35 157	33 726	1 431
8	16 20	15 70	50	33 113	32 091	1 022
9	15 50	15 00	50	31 682	30 660	1 022
10	14 85	14 50	35	30 353	29 638	715
Means	14.6110	14 0715	5395	29 7585	28 6592	1 0993

Power measured by A 3 8358 per cent. greater than measured by B

NOTE.—First series. A on left-hand pipe. Second series A on right-hand pipe

This comparison has no value or meaning unless one instrument is known to be accurate and standard.

To test the friction of the working parts of the indicator, if a means can be secured of obtaining a manageable and variable steam-pressure, try the instrument at various pressures, as shown by a *reliable* steam-gauge, and compare the gauge-readings with those obtained by measurement of the diagram, as exhibited in the figures on page 158.

In *A*, Fig. 28, the diagram is that given by an instrument fitted with a "30-lb. spring," and having considerable friction of pencil movement; in *B*, the diagram is that of an indicator of little friction, and fitted with a "20-lb. spring."

How far such tests and comparisons may be taken as quantitatively gauging the value or accurracy of the instrument is

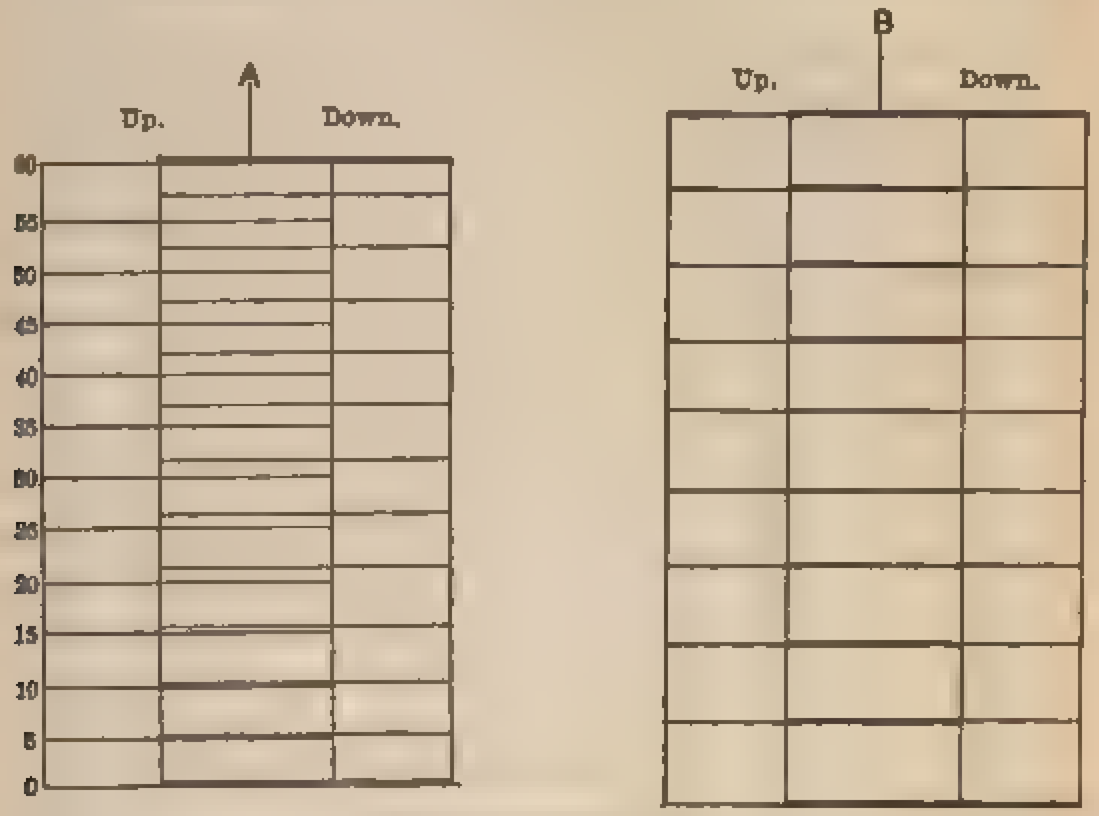

FIG. 28.—INDICATOR TEST

uncertain. Probably the less the friction—as a rule, though not always—the better the indicator; but the action of movement in use, and the effect of inertia of parts, so modify final results, the former by lessening, the latter by exaggerating, the effect of friction, that it is quite impossible, so far as is to-day known, to predicate definite quantitative deductions relating to the accuracy of the instrument. We can only say that the lighter the parts, the less the friction, and the more accurate the spring-tensions and the pencil-movements, the better the indicator, and that the best now made, under the usual working conditions of the best engines, may be expected to give sensibly correct diagrams. This fact does not make it any the less imperative that every indicator to be used in any important work should be fully and carefully tested.

The standardization of the indicator is the more important from the fact that there is no available means of checking its work. The work of the engine, as otherwise customarily measured, is rarely known with accuracy, and the Author has known several indicators, used under similar conditions, to differ among themselves 10 and 15 per cent., with no means at hand of determining which of them were wrong, or the extent of their errors. Where a dynamometric brake is used, the check is more satisfactory, as the friction of the engine is commonly known, or ascertainable within a comparatively small limit of error. The best makers of indicators are, however, usually prepared to guarantee, to standardize, and to give variation-tables of their instruments; and the errors are now reduced in such cases to probably very small amounts.

55. **The Attachment of the Indicator** should always be so effected that its piston may receive precisely the pressure simultaneously acting on the engine-piston, and so that the motion of the paper shall exactly reproduce, as to time and in its proper proportion, the movement of the piston. This means that the steam-connection should be amply large and free from bends and angles, and that the cord and reducing motion giving movement to the paper-cylinder, or -drum, should be so arranged as to lead right, and to be perfectly free from lost motion or stretch.

In attaching the instrument, it is usual to drill a half-inch hole in each end of the steam-cylinder, and to make connections with half-inch pipe to the indicator-cock as directly as possible. In many cases it will be found that the drilling has already been done by the builder of the engine. The opening into the cylinder is commonly in the clearance space back of the piston. Care should be taken that it is not covered by the piston at the end of stroke, and that the in-rush of steam from the steam-port is not likely to produce any sensible effect by blowing across the hole. Especial care should be taken to prevent chips from the drill falling into the cylinder and lodging where they can do injury. The work should, if practicable, be done with the heads removed; if this is not practicable, a

little steam should be turned on and the chips blown out before starting. If the indicator-cock can be screwed directly into the cylinder, it is an advantage. The indicator should, if possible, stand in the vertical position when in use, and one should be placed at each end of the cylinder, and diagrams taken as nearly simultaneously as possible. The cock between the indicator and the cylinder should be of the full size of the pipe, and should be so made that steam may be at any time either turned on the instrument or blown out into the air to clear the passages, and to see that all is right.

The Reducing Motion is made in many ways, and is often, by the ingenious engineer, improvised for the occasion. It must reduce the motion of the piston so as to give a correct throw at the drum and exactly proportionally at every part of the movement.

One of the simplest and best devices is the "Brumbo Pul-

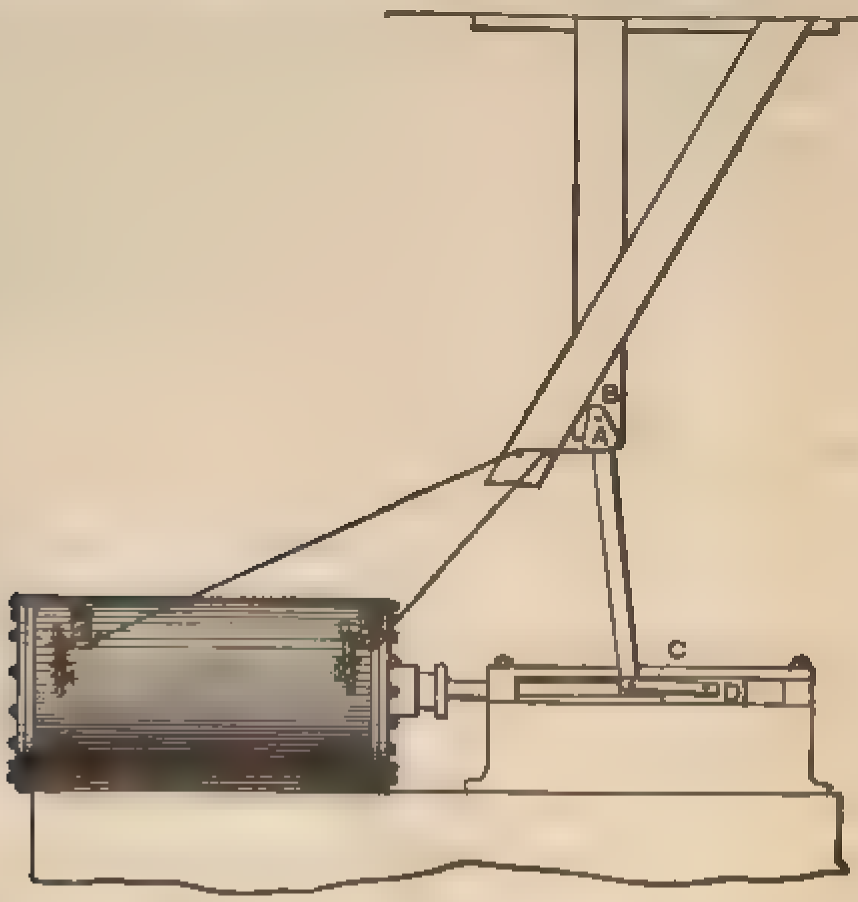

FIG. 29.—THE BRUMBO PULLEY.

ley," Fig. 29. It consists of a sector, *A*, vibrating about an axis, *B*, and actuated by an arm, *C*, and a link, *D*; the latter connected as directly as possible with the cross-head. This may

often be accomplished by attaching its free end to a set-screw on the latter. The sector is usually of wood; but if to be permanently set, is sometimes a light frame of brass. The arm and link are usually of light iron or steel, secured together with nicely fitted pins. The longer the arm in proportion to the stroke of engine-piston, the truer the action; the proportion of two to one should be obtained if it can be done.

The accompanying sketch illustrates a neat device for securing a correct adjustment of the indicator-cord when taking motion from a simple suspended lever. The pin to which the cord is attached is set in a right-angled piece of wood with lines marked upon it parallel to its lower edge and indicating the proper direction of the cord. This is secured on the pendent lever in proper position, when the latter hangs vertically, the engine at mid-stroke, and is then fastened securely by small screws.

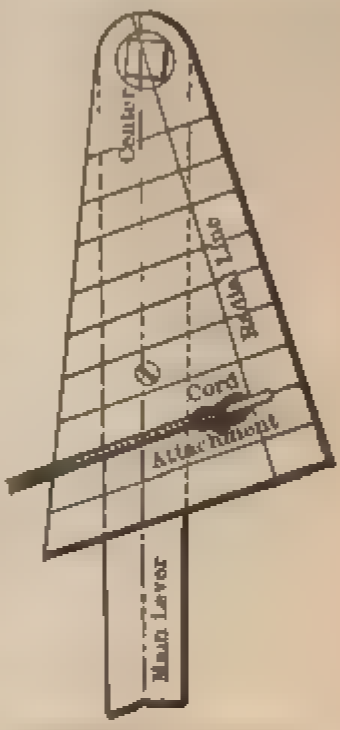

Fig. 30.—Cord Attachment.

A modification of the Brumbo arrangement which the Author has found to work excellently well on high-speed engines is illustrated in the next sketch, as designed originally by Mr. Sweet. When used under the direction of the Author in work done at the Sibley College of Cornell University, it was constructed as follows:

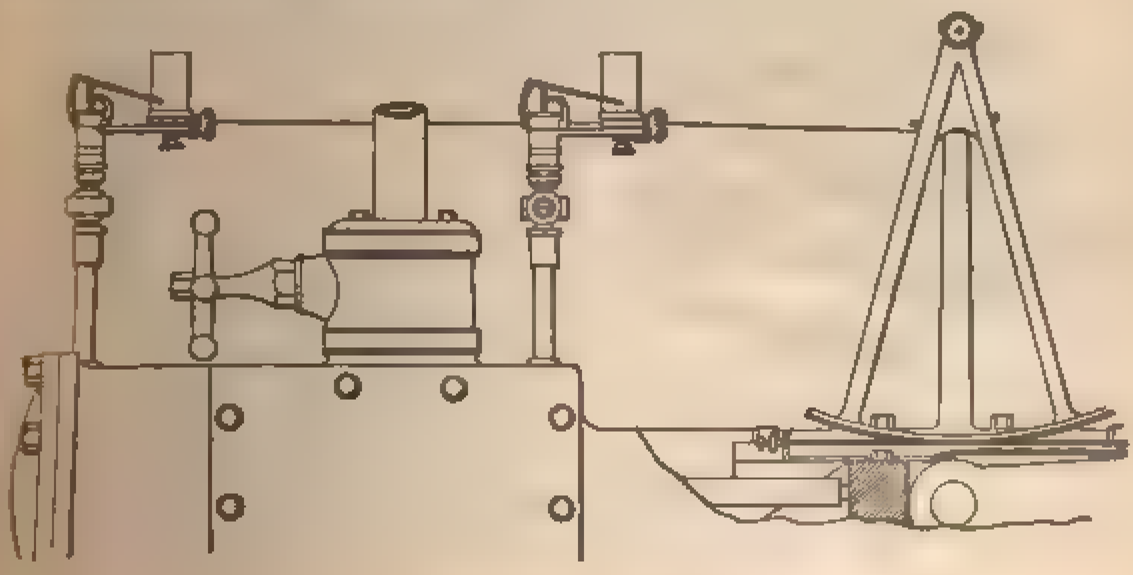
Fig. 31.—Indicator Attachment

The reducing mechanism used in connecting the indicator-barrel to the cross-head of the engine was fitted with a very firm connecting arrangement, and with an ingenious detaching device. A sector was constructed which was pivoted above the cross-head and hung in the vertical plane above the latter, the engine being horizontal. The arc of the sector carried a pair of steel ribbons, one attached to each end, each carried around the arc and secured, at its opposite end, to the end of a bar fastened on the cross-head, in such manner that, the two ends of the ribbons at the cross-head bar being well secured and tightly drawn up by means of screws placed conveniently for the purpose, all back-lash was prevented, and an absolutely exact synchronism of movement of indicator-line and cross-head was obtained. A smaller sector at the upper part of the larger one was the carrier of the cord, and the combination was thus a perfect means of reproducing the motion of the engine on the smaller scale required in working the paper-barrel of the indicator. The "cord" was piano-wire, a material much less liable to cause difficulty by stretching than any other that was available. Its free part was kept taut by a "spiral" (helical) spring, attached beyond the point of connection with the paper-cylinder.

The cord may either be taken around a groove in the rim of the sector or led from a properly set pin on its side. The latter is the more accurate, provided the cord at half-stroke is led off at right angles to a radius of the pulley passing through the centre of the pin.

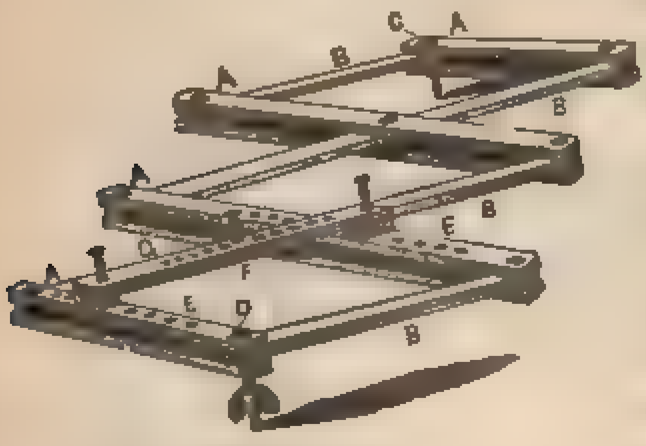

FIG. 32.—THE PANTAGRAPH.

A "pantagraph motion," if well made, nicely adjusted, and properly attached, makes an excellent reducing arrangement. This is seen in Fig. 32.* It consists of a system of levers of wood; those marked *B* are single strips, and

* American Machinist Dec. 27, 1879.

those marked *A* double strips. The pivot-holes should be bushed The strip *G* should be arranged so that it may be shifted in the holes *E*, and bring a hitch-pole, *F*, in a line passing through pivots *C*, *D*. The end pivots *C* and *D* should have a projection below, with the end somewhat pointed. The engine cross-head must have a vertical hole in it somewhere, so that pivot *C* can be dropped into it. A stake must be set in the floor near the guides, having a socket for the pivot *D* in its top. Its socket must be level with the cross-head socket, and must be directly opposite the former when the latter is at mid-stroke. The indicator-cord is hooked to the centre peg *F*, and the cord should lead off parallel with the guides.

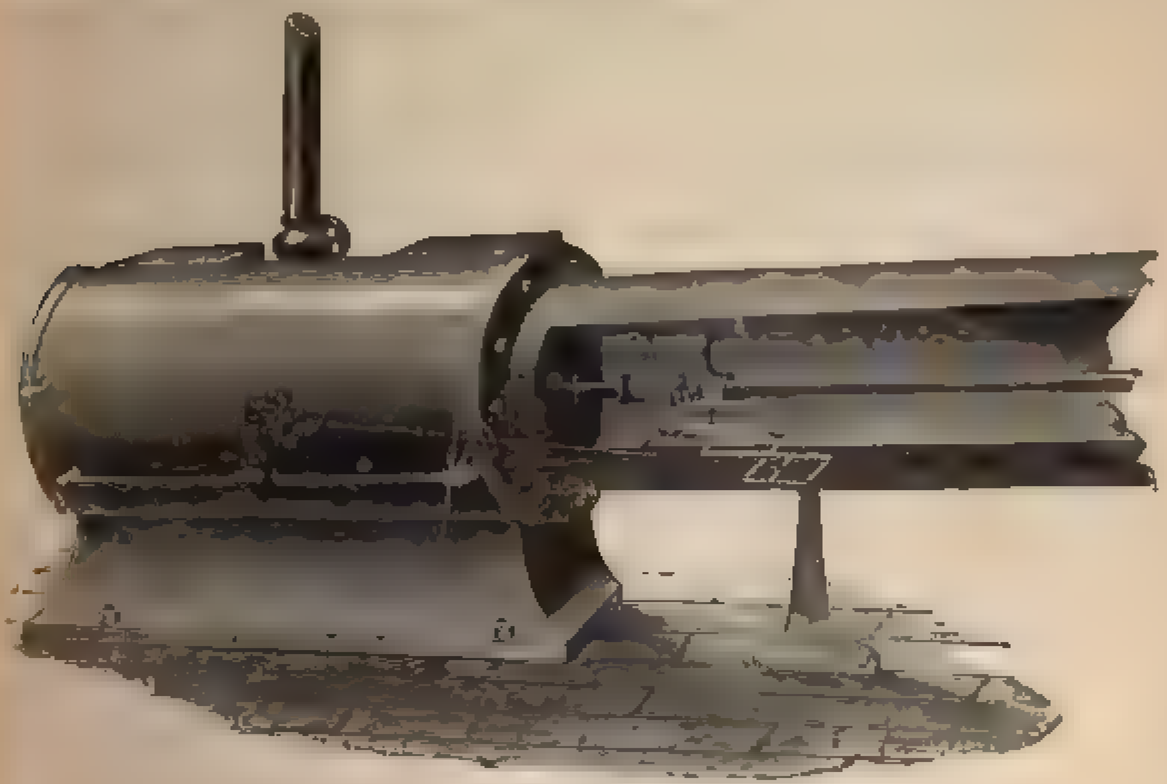

Fig. 33.—The Pantagraph in Place.

The next illustration shows the apparatus in place, and the indicator attached. Various modifications of this device are in use, all of which embody the same principles.

Fig. 34 shows another form of pantagraph. The working end, *A*, takes motion from the cross-head, and *B* is attached to the floor. The pin, *D*, is fixed in line between the pivot and the working end, and the pulleys, *E*, guide the cords.*

* Barrus on the Indicator.

Avoid the use of long cords. If the motion must be carried a long distance, strips of wood may often be arranged in their place and operated with direct connections. Braided

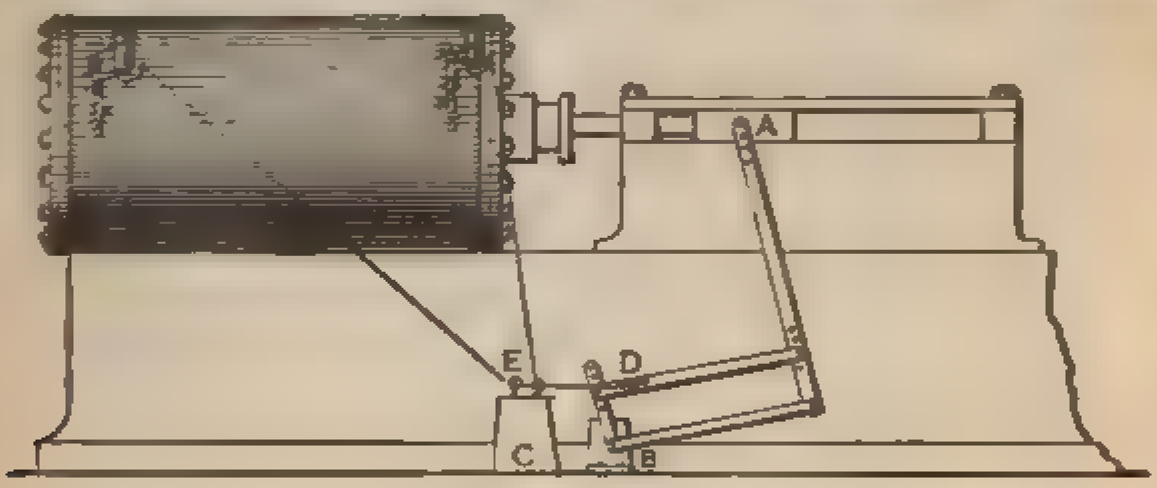

FIG. 34.—THE PANTAGRAPH.

linen cord, a little over one-sixteenth of an inch in diameter, is a suitable material.

The next two engravings exhibit Mr. Thompson's usual methods of attachment for engines of his own design: *A* is the

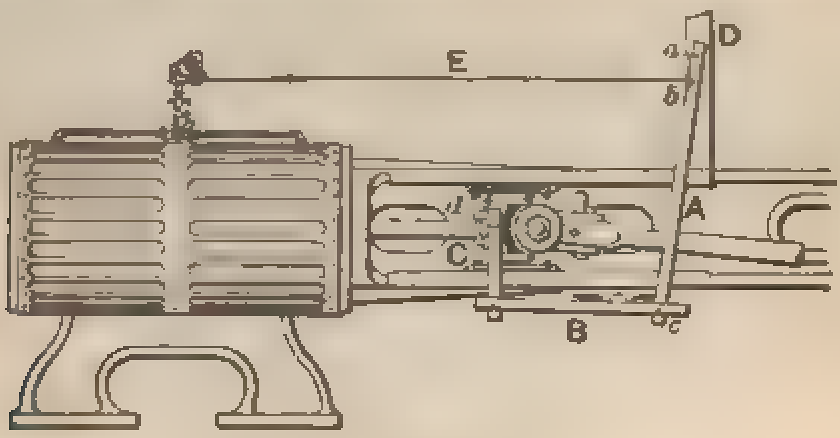

FIG. 35.—INDICATOR "RIG."

lever, *B* the connecting bar, *C* a strip of board attached to the cross-head, *D* a firm support, and *E* the indicator-cord, which is shown horizontal. This horizontal direction allows the pivot *a*, the cord pin *b*, and the pivot *c* to be in line, and when no pipe fittings are used to connect the instrument, and it consequently is shifted from end to end of the cylinder, it is correct for both positions.

Fig. 36 shows a device that will give a movement perfectly

free from distortion. The cord is attached to the end of a short bar, which slides freely in a bearing in the carrying-post. This bar

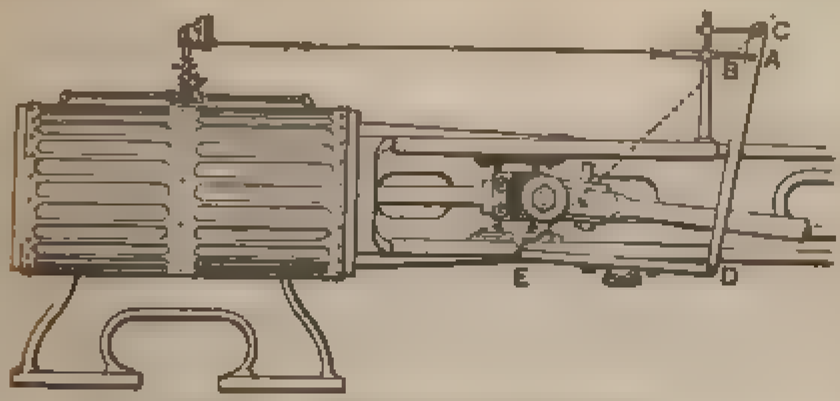

Fig. 36.—Indicator Attachments.

is connected to the lever *CD* by a link *AB*. The lever is connected to the cross-head at *E* by a bar, *DE*. The pivots *C*, *B*, *E* are in line at all times; and the distortions of the movement of the lever due to the vibration of *DE*, will be corrected by the equal vibration of the link *AB*; since, $CD : DE :: CA : AB$. This, to be correct, must be proportioned for the engine. The cord should be nearly level.

The cord employed should be as short as possible. If unavoidably long, a fine wire of steel, of iron piano wire, or of hard-drawn brass should be used. Braided cord is usually supplied by makers of indicators which has been made especially for the purpose and well stretched. A hook on the end of the cord attached to the drum and a loop, Fig. 37, on the adjacent

Fig. 37.—The Loop.

end of the cord from the reducing motion afford means of ready connection and disconnection. The loop is adjusted, before hooking on, to just the right length so as to avoid liability of accident by maladjustment when starting. The spring chosen should usually be rated at above one-half the maximum gauge reading; in other words, so that the maximum rise of the pencil may not be above two inches. The minor details of op-

eration are always fully described in the instructions supplied by the maker of the indicator.

Mr. Lyne's method of attachment of the indicator to the locomotive is shown in the accompanying engraving.*

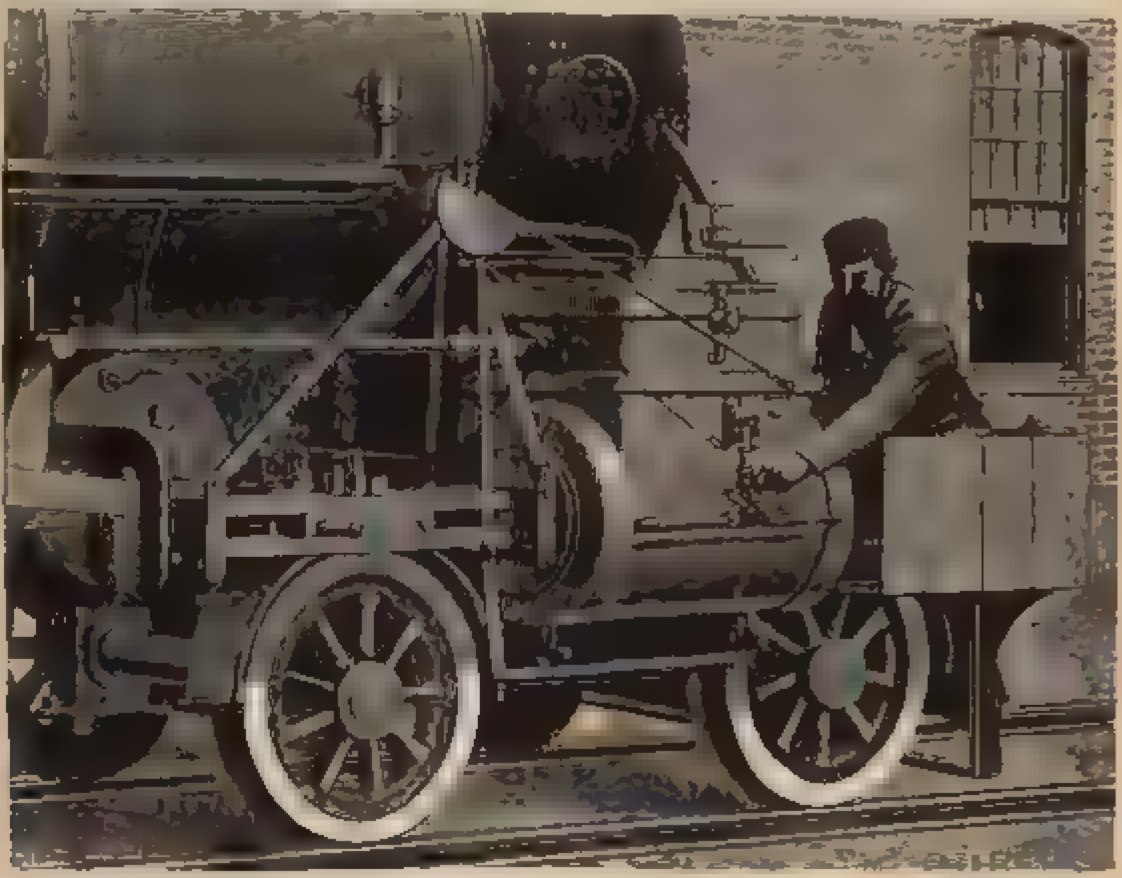

FIG. 38.—"INDICATING" LOCOMOTIVES.

In making connections, a hole is drilled close to the flange at each end and tapped to fit a ¾-inch nipple (wrought-iron pipe). These holes should be so drilled that the piston may not cover the holes at the ends of the stroke. After the holes have been drilled and tapped (which may be done without removing the cylinder casing or back cover), two nipples having ¾-inch malleable-iron elbows upon the ends are screwed in place. These should be tightly screwed up, and set looking towards each other. Two long nipples, having brass collars brazed upon one end, are then cut of a proper length to fill the space between the elbows and cock. Brass should be used for the collars, as solder will not run through the threads if the

* Am. Machinist, Apr. 1, 1882, p. 1.

collars are of iron. The end of one of these nipples is to be cut in the usual way and tightly screwed in place, while upon the other nipple the die is run up 1½ inch for the application of a lock-nut. It is then screwed into the elbow, the three-way cock is put in place, the joints being made with washers of annealed sheet copper. Rubber and all other fibrous material should be avoided in making these joints. The nipple having the lock-nut is screwed out from the elbow until the joints are firmly made upon the cock, then the lock-nut is to be set up against the face of the elbow, a few strands of lamp-wick having previously been wound between the two surfaces to avoid any possibility of leaks. The ends of all the pipes should be carefully rounded inside with a file, and nothing but clean oil should be used in making the joints. If red lead is used, the operator, in taking the diagrams, will have the annoyance of seeing his instrument stick, and will be obliged to remove the piston frequently and pick the lead and dirt out of the instrument.

The three-way cock, shown in the sketch, is of a heavy pattern. The brass centre of the handle has a wooden covering, which protects the hands from being burned. The passages are of quite easy curvature and of large diameter—$\frac{7}{16}$ inch in this case.

The details of this construction are seen in the sketches herewith given.

In applying these devices to a locomotive, place the cross-head at half-stroke, attach the frame to the guides, and set the lever at right angles to the guides and screw up all the nuts, being careful to spread the blocks so that the cross-head will not strike them. Adjust the lever vertically, in the following manner: With a pair of trammels set to the length of the lever, describe an arc, and draw a straight line with the ends touching the arc. Measure from the centre of the line to the arc, and set the centre of the lower pin in the lever half this distance below the centre of the cross-head, then tighten the nut on the upper pin. This will equalize the vibration.

This should all be done beforehand, and the distances put

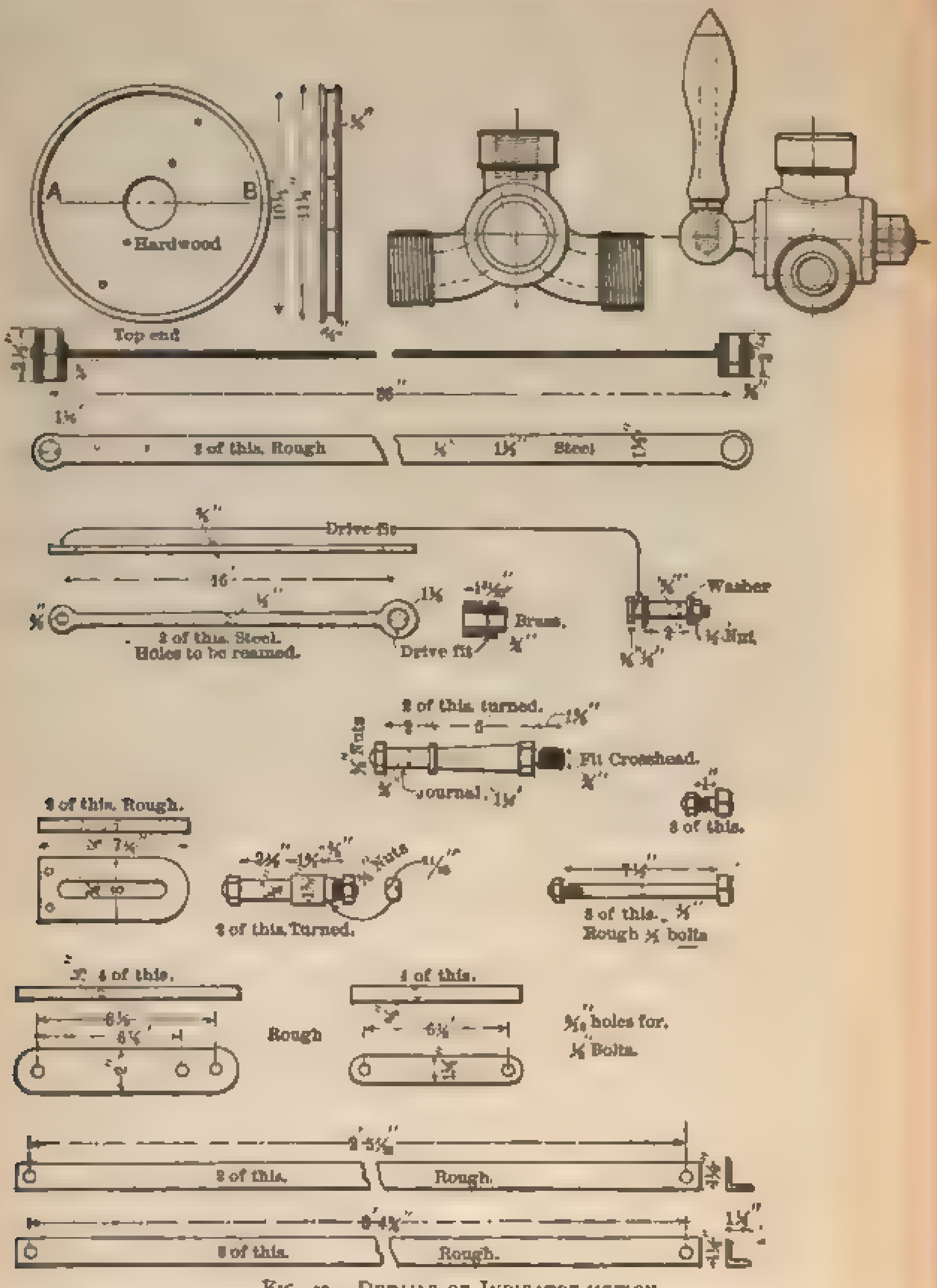

FIG. 39.—DETAILS OF INDICATOR-MOTION.

down in a book for reference. To prevent the cord from getting out of the groove, a wire loop should be used, as shown. This arrangement can be applied and adjusted with-

out moving the engine. The indicator-cord is short, and the lever quite long. The cord is best braided linen line, well stretched and lightly waxed. The box bottom is 9 inches below the beam by a ⅝-inch bolt through it and the flag-stand; the back is supported by an iron bracket attached to the stud in the centre of the cylinder-head.

The counter is attached to a board bolted to the front brace, as shown.

In applying an indicator to the steam-chest, drill a hole in the centre horizontally and vertically; then screw in a half-inch nipple and elbow, and set up the indicator in line with the grooved arc. The cord connection will be very short.

Holes should be drilled in every cylinder while the engine is in the shop, and brass plugs, with hexagon heads, screwed into them. The cross-heads should be drilled and tapped, so that upon an hour's notice the indicator may be attached without the necessity of doing any work. In cold weather it is desirable to erect a screen to protect the operator from the wind.

The method of attachment found, on the whole, most convenient, in the work of the Author on vertical marine engines, is that shown in Figs. 40 and 41.* This apparatus was designed for the Author by Mr. Lyne, and used on the steam-yacht "Namouna," while preparing to make some improvements. The engines are of the compound "tandem" type, their cranks at right angles. The high-pressure cylinders are 22 inches in diameter and placed above the low-pressure cylinders, which are 42 inches in diameter with a stroke of 28 inches. The propeller has four blades and a pitch of 18 feet; the boiler-pressure is 80 or 85 pounds per square inch; the engines make from 80 to 85 revolutions per minute. At *A* is the high-presssure cylinder; the low-pressure at *B*; while *C* is the frame. *E* the guides, *D* one of the columns supporting the engine. A wrought-iron arm, *G*, is bolted to the pin on the cross-head *F*. This arm had a rectangular end for the slide *K*. The arm *G* was at right angles to the guides, and, by the aid of a steel steady pin, can be readily removed and replaced, the

* Am. Machinist, Aug. 19, 1882, p. 3.

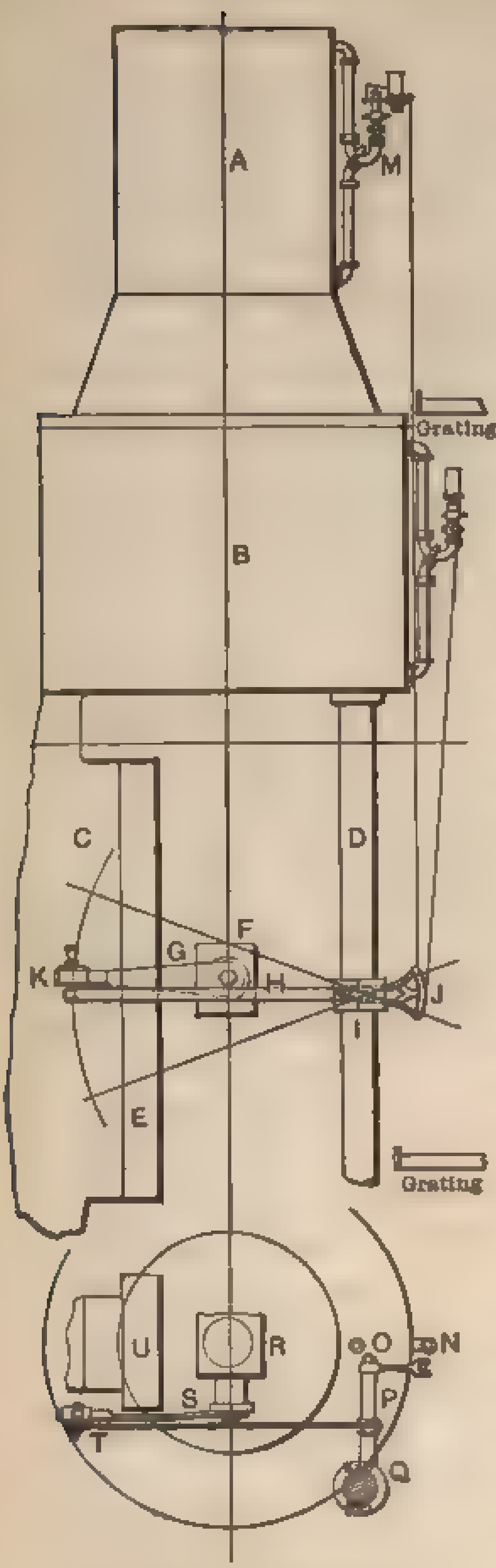

FIG 40.—INDICATOR MOUNTING.

hole for the pin being reamed tapering and the body of the bolt filling the hole. The principal object in making this arm so long was to use a lever, *H*, 40 inches long. Errors are less with a long lever than with a short one. The lever was attached to a thimble, *P*, and a pin or feather inserted to avoid possibility of the lever changing its position. A collar, *I*, was fitted to the column *D* by being bored with a piece of iron $\frac{1}{16}$ inch thick in the joint, so that, after the collar was finished and this iron removed, the collar would grip the column.

A segment, *J*, was bored to fit upon the thimble *P* in plan, and to the cross-head; *T* is the slide, *Q* the collar attached to the column, and *P* the thimble with grooved segment. *N*, *O* are the positions in plan of the two indicators *L*, *M*.

The pipes were all of brass, and neatly finished. The cords run directly to the indicators, and no guide-pulleys are used. The grooved segment has a radius to give a diagram 5 inches long.

The advantages are as follows: It may be run constantly

with but little wear, as the wearing surfaces are all large, and it is always ready for use. It is simple and easily made, and diagrams may be taken in a heavy sea with as great accuracy as in smooth water, as there are no guide-pulleys attached to the woodwork of the vessel.

Two cords for working the indicators are attached to the grooved segment *J*, by passing the end of each through a hole at each side of the groove, as shown at *a a a* in plan, and knotting the ends.

The steam-pipes to the indicators are $\frac{3}{4}$ inch in diameter.

The experience of the Author with this arrangement was thoroughly satisfactory. It is somewhat costly, in comparison with less perfect devices; but its operation is so effective as to fully compensate that disadvantage where, as in this case, it is intended to be a permanent attachment, and kept ready for daily use.

The working drawings of the details of this attachment are presented in the next figure.

A is the lever, *B* the composition sleeve, upon which is fitted the segment *C*. The bearing *D* is turned down in the middle to form an oil-chamber. The bearing *D* is screwed into the collar *K*. The screws for holding this collar together are operated by a screw-driver. The slide *F* is made of composition, having a gib to take up the lost motion. No set-screw is used, as it is safer to insert a liner on top of the gib.

The pin *J* is attached to the slide *F*, and forms a journal for the lever. The thimble *I* is fitted to the pin *J*. *GH* shows the arm bolted to the cross-head.

The following are the details of proportioning several simple indicator-motions devised and described by Mr. Nystrom.*

Fig. 42 represents one of these indicator-motions; though not absolutely correct, the error is insensible. It consists of a horizontal lever, *L*, with its fulcrum at *C*, and the other end attached to the pendulum *P*, the lower end of which is attached to the cross-head. The fulcrum *C* should, as advised by its designer, be placed near the indicator *I*, so as to make the

* Mechanics, June 1883.

lever L of nearly the same length as that of the cord conr ing the indicator with the pendulum. The error on the will then be the difference of the versed-sines of half the an

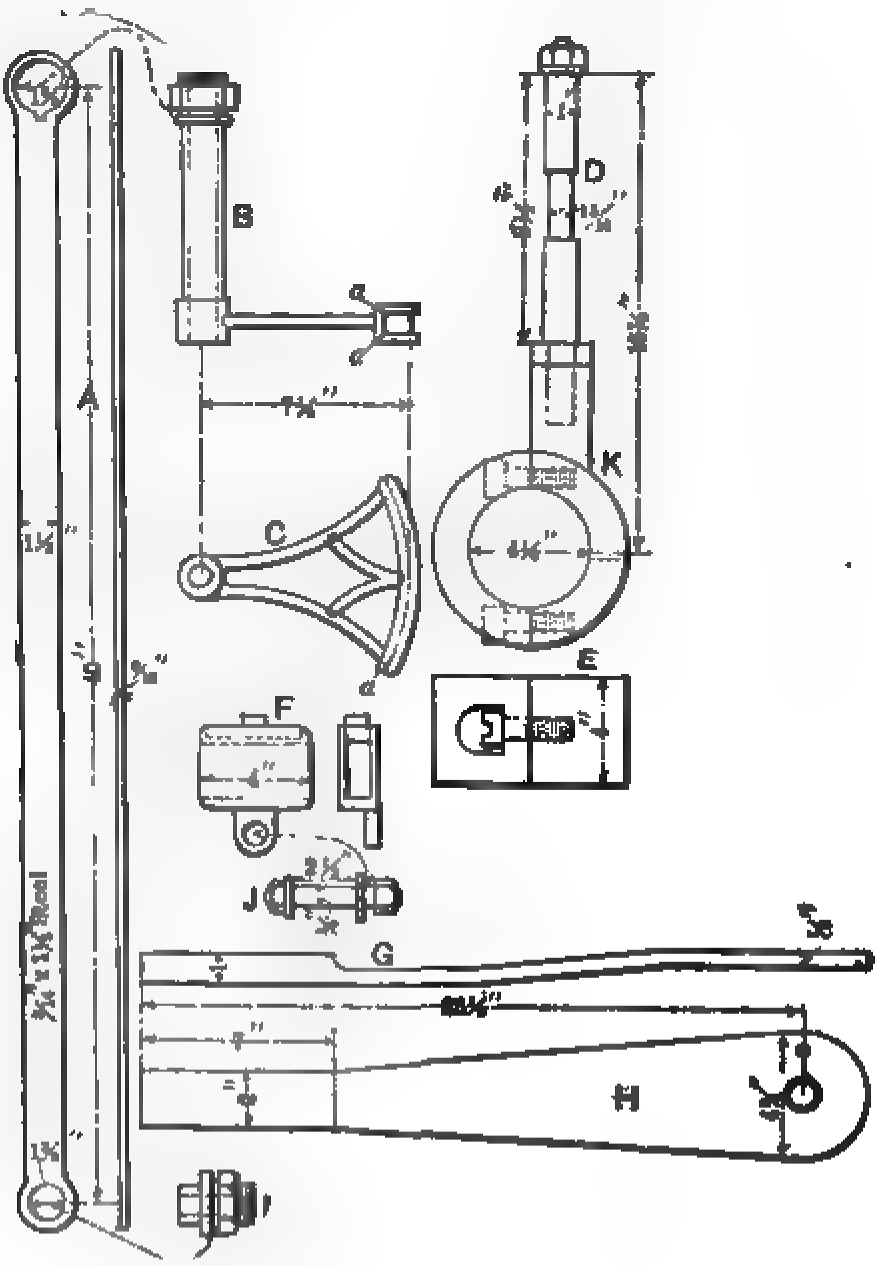

FIG. 41.—DETAILS OF INDICATOR GEAR.

formed by the motions of the lever and cord. All li dimensions in the following analysis are to be expresse inches.

L = length of the lever;
l = length of the cord from the indicator to the ver position of the pendulum;
P = length of the pendulum;
s = half the stroke of the steam-piston;
ϕ = angle moved by the lever L;

ϕ^1 = angle moved by the cord;
δ = differential;
v = half the angle of the pendulum motion;
e = vertical motion of the joint of the pendulum and lever.

$$e = P - \sqrt{P^2 - s^2}.$$

Example.—The pendulum $P = 36$, and $s = 12$ inches. Required the motion of the joint.

$$e = 36 - \sqrt{36^2 - 12^2} = 2.0588 \text{ inches.}$$

Half of this motion will then be 1.0294 inches.

Assume the lever $L = 48$ inches, and half the angle ϕ will then be

$$\sin \tfrac{1}{2}\phi = \frac{1.0294}{48} = 0.021445 = \sin 1^\circ\ 13'\ 40''.$$

The versed-sine for this angle is $0.00023 \times 48 = 0.011$ of an inch.

$h = 35.5$ inches vertical mean height of the cord above the direction of the stroke s; then the versed-sine will be reduced to $\frac{h}{P}$, or $\frac{0.011 \times 31.5}{36} = 0.009625$ of an inch.

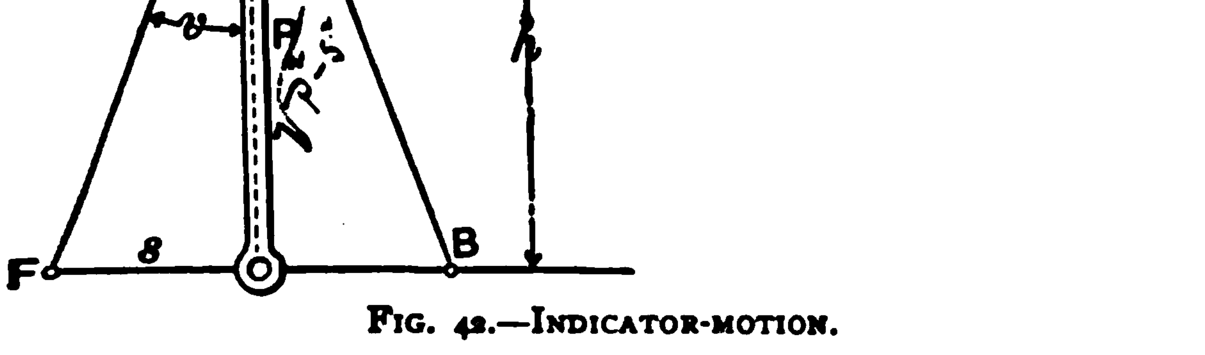

FIG. 42.—INDICATOR-MOTION.

The vertical motion of the cord at the pendulum will be $\frac{2.0588 \times 31.5}{36} = 1.80145$ inches, of which one-half = 0.900725. Half the angle of the cord will then be

$$\sin \tfrac{1}{2}\phi' = \frac{0.900725}{48} = 0.00187765 = \sin 1^\circ\ 4'\ 30''.$$

$$l = 54 \text{ inches.}$$

The versed-sine for this angle is $0.00017 \times 54 = 0.00918$ of

an inch. The error on the card caused by this indicator-motion will then be $0.009625 - 0.00918 = 0.000445$ of an inch. This error is too small for detection on the indicator-card, but it can be removed entirely by placing the fulcrum C at C' on the other side of the indicator. Fig. 42 can therefore be considered a reliable indicator-motion.

The letters F and B represent the position of the pendulum when the piston is at the front or back of the cylinder.

Fig. 43 represents another of these indicator-motions for the locomotive. It is similar to that of Fig. 42, except that the lever L is placed at the other side of the pendulum, as a matter of convenience. There were two cylinders to be indicated simultaneously, for which purpose horns, ab, Fig. 44, were fixed on the pendulum from which cords were led. The pendulum was of steel, 2 inches wide by ⅝ inch thick,

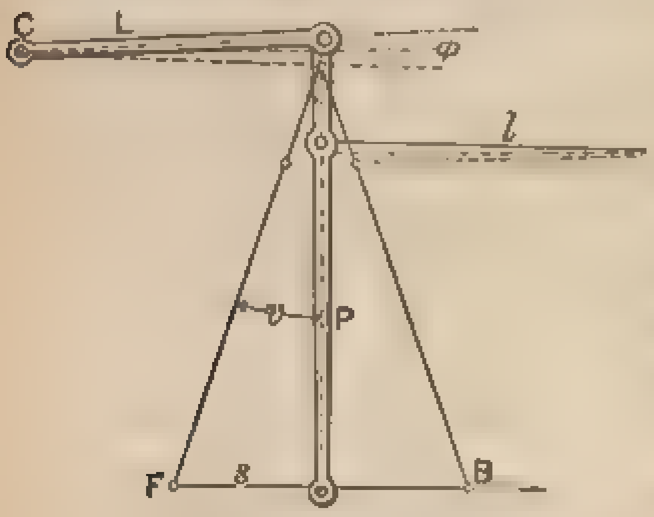

FIG. 43.—INDICATOR-MOTION.

and the pivot-holes were ½ inch. The ends of the pendulum were made 3 inches in diameter for the purpose of making it firm against uneven action on the horns, which were ¾ inch in diameter at the pendulum and tapering to ½ inch at the eyes. This made a very rigid system, which worked well. The fulcrum C for the lever L was fixed on the foot-board of the locomotive. The pendulum was 36 inches, lever 24 inches; and stroke of steam-piston 24 inches, stroke of card 3 inches. The calculation for error is the same as that for Fig. 42, except that the versed-sine of the cord must be added to that of the lever L. The error so obtained was 0.018, which, divided at each end of the card, makes it 0.009. This error exists at ⅛ from each end of the card, positive at one end and nega-

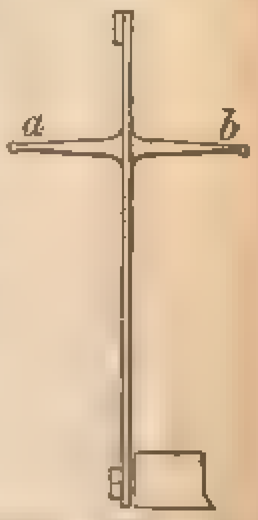

FIG. 44.—HORNS ON INDICATOR-MOTION.

tive on the other; but when cards are taken from both ends of the cylinder the errors compensate.

The angle v of the pendulum at the ends of the stroke is

$$\sin v = \frac{s}{P} = \frac{12}{36} = 0.3333 = \sin 19^\circ\ 28'.$$

The indicator-motion, as represented by Figs. 45, 46, and 47, is a defective indicator-motion.

Fig. 45 exhibits the characteristics of link indicator-motions. The cross-head moves in the direction of the dotted line FB; the link L is made very short in proportion to the length of the pendulum P, for the purpose of better illustrating the motion. The different positions of the link and pendulum are numbered 1, 2, 3, etc., of which in the first position the link and the

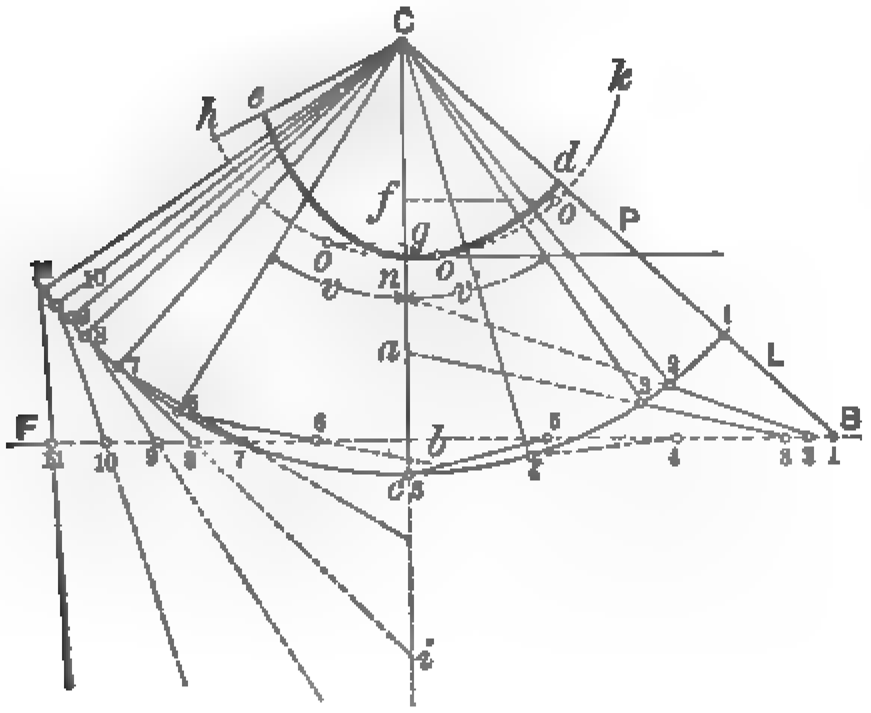

FIG. 45.—LINK INDICATOR-MOTION.

pendulum are in a straight line; the cross-head is stationary while the pendulum moves. The line Cc is the vertical position of the pendulum. The point on the vertical Cc, where the direction of the link crosses, shows the motions of the cross-head and of the pendulum; the motion of cross-head is to pendulum as Ca is to Cc. In the fifth position the link

crosses the vertical at c; the motions of the pendulum and cross-head are alike. In the eighth position the line of the link crosses the vertical at i; the motion of cross-head is to pendulum as Ci is to Cc.

On the right side of the vertical the pendulum moves faster than the cross-head, and on the left side the cross-head moves the faster. When the link is less than half the length of the pendulum, the latter should move over a much smaller angle on the link side than on the other side of the vertical. The motion of the pendulum transmitted to the indicator by a circle-sector, h_1, n_1, k_1, will make the cord move too fast at the ends of the stroke. A sector should not be circular, but of the form e_1, n_1, d_1 for the proportion of link and pendulum shown in Fig. 45; then,

$$C_1 e_1 : C_1 n_1 = C_1 a_1 : C_1 c_1$$
$$C_1 d_1 : C_1 n_1 = C_1 b_1 : C_1 c_1$$
$$C_1 e_1 : C_1 d_1 = C_1 a_1 : C_1 b.$$

The indicator-cord fixed on the pendulum, without a sector, will move too fast on the link side and too slow on the other side at the end of stroke. The cord should be fixed at o, on the link side of the vertical, and so that

$$C_1 f_1 : C_1 n_1 = C_1 a_1 : C_1 c_1$$
$$C_1 g_1 : C_1 n_1 = C_1 b_1 : C_1 c_1$$
$$C_1 f_1 : C_1 g_1 = C_1 a_1 : C_1 b.$$

When the cord is attached to the sector h_1, n_1, k_1, or to the pendulum at n_1, the pendulum should make a smaller angle on the link side and a greater angle on the other side, but it will ~~not~~ make a true card in any case.

Fig. 46 represents a motion very much used. The direction of motion from the cross-head bisects the versed-sine of the angle made by the pendulum, and the angles v are alike on both sides of the vertical.

Here,

P = 18 inches;
L = 8 inches;
e = 4 inches;
h = $17\frac{1}{8}$ inches;
v = 26° 23′;
S = 16 inches;
Ca = 15.05 inches.

From these data we find that the indicator will move 14 per cent. faster at the back than at the front end of stroke.

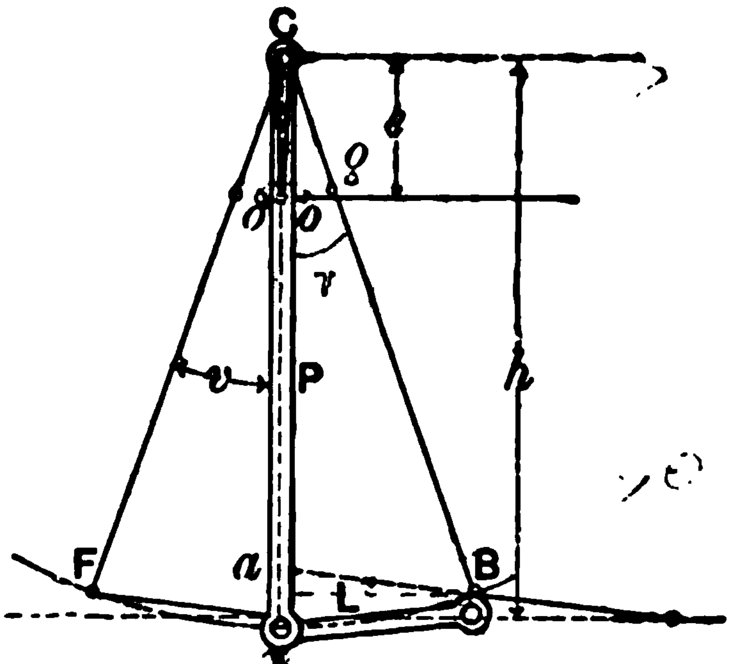

FIG. 46.—INDICATOR-MOTION.

Lengthening the link, correct the defect in part, and the motion would make a tolerably correct card.

Sin v = S/2P

Fig. 47 represents a motion in which

P = $48\frac{3}{16}$ inches, length of pendulum;
L = $34\frac{11}{16}$ inches, length of link;
h = 38 inches, height of fulcrum;
v = 30°, half angle of motion;
S = 48 inches, stroke of piston;
Ca = $44\frac{1}{4}$ inches;
Cb = 39 inches.

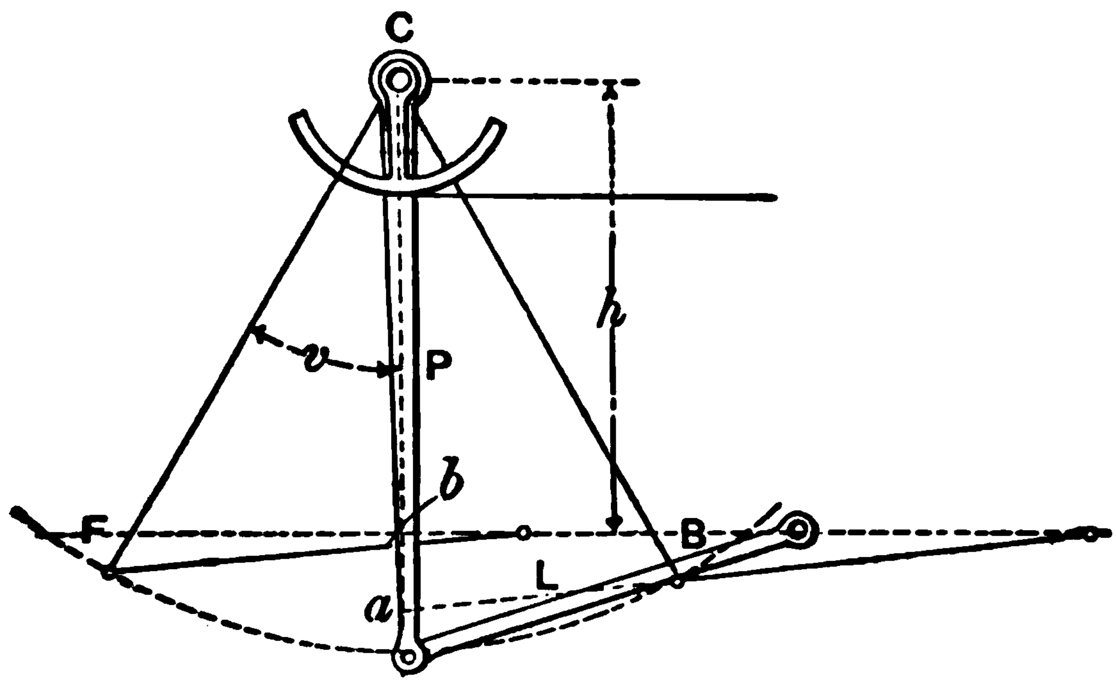

FIG. 47.—INDICATOR-MOTION.

The indicator would move nearly 14 per cent. faster at the front than at the back end of stroke. The link should be

made 29 inches long, to make the angle of motion greater on the link than on the other side the vertical. The motion would then produce a better card.

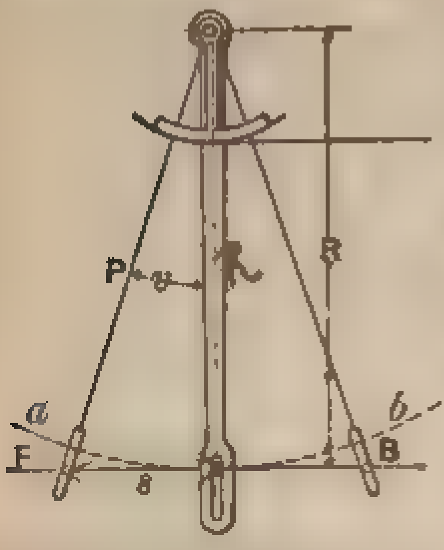

Fig. 48.—Slot Indicator-motion.

Slot motions for indicators have been tried, and abandoned.

Fig. 48 represents a motion in which the cord is moved by a circle-sector.

The slot-pin is fixed in the cross-head of the engine, and the problem is to find the relative motion of pin to slot on the arc ab. The height R of the fulcrum is constant.

$P =$ distance from the fulcrum to any position of the slot-pin;

$S =$ half the stroke of the steam-piston;

$a =$ half the length of the arc ab.

$$\tan v = \frac{R \sin v}{\cos v}.$$

Differentiate this formula, and we have

$$\delta \tan = \frac{\cos \delta \sin - \sin \delta \cos}{\cos^2},$$

$$\delta \tan = \frac{(\sin^2 + \cos^2)\, \delta a}{\cos^2},$$

$$\delta \tan = \frac{\delta a}{\cos^2},$$

and $$\delta a = \cos^2 \delta \tan.$$

The differential of the tangent is the differential motion of the pin, which we may take as unit,

$$\delta a = \cos^2.$$

The motion of pin is to arc motion as 1 : $\cos^2$.

If radius $R = 36$ inches, and $s = 15$ inches, v will be

$$\tan v = \frac{s}{R} = \frac{15}{36} = 0.41666 = \tan 32^\circ\ 37'.$$

The cosine for this angle is 0.9231.

The square $0.9231^2 = 0.8521$, and when the motion of the pin is 1, near the end of the stroke, that in the arc at a will be 0.85, 15 per cent. too slow.

The error may be corrected by converting the sector into a triangle, as shown by the dotted line.

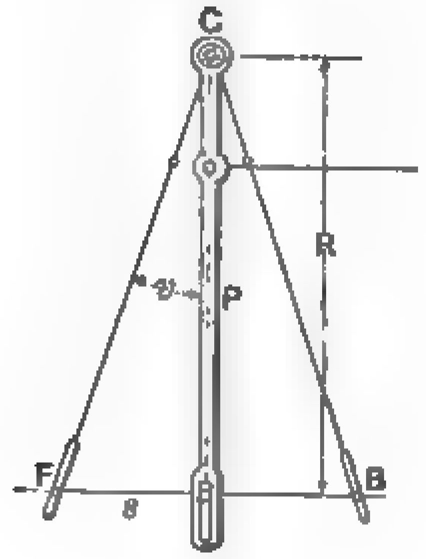

FIG. 49.—SLOT MOTION.

Fig. 49 represents a motion in which the cord is fixed without a sector. In this case δ sin must be inserted for δa; then

$$\delta \sin = \cos \delta a.$$

$$\delta a = \frac{\delta \sin}{\cos}.$$

Insert this for δa, and we have

$$\frac{\delta \sin}{\cos} = \cos^2, \text{ or } \delta \sin = \cos^3.$$

The motion of pin is to cord as $1 : \cos^3$. Let $R = 36$, $s = 15$, $v = 22° \ 37'$ and $\cos v = 0.9231$; $0.9231^3 = 0.7865$. When the motion of the pin is 1, near the end of the stroke, that of the cord will be then 21 per cent too slow. Slot motions distort the card at the ends of stroke, where greatest accuracy is required for exhibiting the method of distribution of steam.

56. Precautions essential to the successful employment of the indicator have been already detailed at some length; briefly summarized, they are: (1) Make sure of accuracy of construction of the instrument in its dimensions and fitting; (2) Secure exactness in scale of the spring employed, not when cold, but when hot and in use; (3) Demand the utmost lightness and stiffness of moving parts; (4) See the spring in the paper-drum correctly adjusted to the speed of the engine; (5) See that the instrument is well lubricated, and with the best of light oils, and that it works freely and without friction; (6) Make the steam-connections short, straight, and large; (7) Use a short cord, and substitute wire where any considerable length is necessary; (8) See that the reducing motion is perfectly accurate, free

from lost motion, and both strong and light; (9) In taking the diagrams, see the steam-cock opened full, the indicator well heated up, the steam condensed in the connections completely blown out, and the touch of the pencil on the paper as light as is consistent with making perfectly legible diagrams.

Every maker gives detailed instructions for care of the instrument and for its dissection and assemblage. The principal points are the following, details varying with the style of the indicator:

Before using any indicator, take it apart, clean, and oil it. Try each part separately. See if it works smoothly; if so, put it together without the spring. Lift the pencil lever, and let it fall; if perfectly free, put in the spring, and connect. Give it steam, but do not attempt to take a card until it blows dry steam through the relief openings. If the oil from the engine gums the indicator, take it off and clean it.

Never use red or white lead in connecting, as it is liable to get into the instrument. Attach the indicator to the cock by coupling the differential threads of the indicator shank and cock. The lighter the spring used, the higher will be the diagram produced, and the more accurate the measurements obtained; in selecting a spring, choose one to give diagram about two inches high.

After the desired number of diagrams have been taken, remove the piston, spring, etc., from the indicator, while it is still upon the cylinder; allow the steam to blow for a moment through the indicator cylinder; then examine piston, spring, and all movable parts, which must be thoroughly wiped, oiled, and cleaned. Particular attention should be paid to the springs, as their accuracy will be impaired if they are allowed to rust; and great care should be exercised that no gritty substance be introduced, to cut the cylinder or the piston. The springs should not be left in the indicator. The pencils can be best sharpened with a fine file.

Each blank indicator-card usually has printed on its back a set of data to be filled out, such, for example, as the following. The number and character of these items differ in the practice

of different engineers; but the more important are never omitted.

Date	Kind of Engine ....
Diagram from ....Engine	..End ....No........
Diameter of Cylinder and Area	Built by .... H. P. Factor....
" " Rod and Area	Initial Pressure ....I. H. P....
Length of Stroke	Brake H. P. ....
Clearance ... .and	Barometer reads ....
Revolutions per Minute	Thermometer ....
Piston Speed	... .
Pressure of Steam, in lbs., in Boiler	....
Point of Cut-off	....
Position of Throttle-valve	Observer ....
Vacuum per Gauge, in inches	*Remarks.*
Temperature of Hot-well	....
" " Injection	....
Scale of Spring ....M. E. P...........	....
Inside Diameter of Feed-pipe	....
" " " Exhaust-pipe	....
....................Valves	....

The Author has used the next form many years. It is amply complete for most cases; indeed is rarely entirely filled out.

Time.........Date.........	R. H. THURSTON, CONSULTING ENGINEER.	Builder of Eng. { .... / ....
Diam. of Cylinder..........		
Length of Stroke	——	Kind of Valve Motion............
Revolution per min........	*Owner of Engine:*	" " Steam-valves .. ..
Speed of Piston...........	....	" " Exhaust-valves...........
Diam. Piston-rod..........	....	" " Condenser ....
Area Steam-port.	*Kind of Work Driven by Engine:*	" " Heater....................
" Exhaust-port		" " Boiler.....................
Piston Clearance..........	....	" " Fuel ....
Port "	....	Temperature of Feed-water.......
Boiler Pressure	.... *Remarks:*	" " Hot-well..........
Initial "	....	Water per hour. ....
M. E. Pressure	Barometer	Coal " " ....

CHAPTER V.

INDICATOR-DIAGRAMS INTERPRETED.

57. Indicator-diagrams taken under proper conditions and with good instruments are diagrams of energy on which the ordinates measure the varying pressures in the cylinder, corresponding to the positions of the piston as measured off by the simultaneous abscissas of the diagram; while the area represents the work done by the steam on the piston of the engine. The forms and relations of the several lines of which the diagram is composed reveal the method of action of the valves and the effectiveness of the pipes and passages as conduits for the entering and the exhausted steam. The correct interpretation of the diagram thus becomes an exceedingly important matter.

58. The Typical Diagram and its Nomenclature, assuming the indicator applied to the steam-engine, are as below.

The curves described on the indicator-cards of engines present many differences as to the mode in which pressure and volume vary, and their figures cannot be expressed by any mathematical formulæ; since it is impossible to separate those irregularities which arise from fluctuations in the pressure of the steam from those which arise from the friction and inertia of the moving parts of the indicator, and also because the law of such changes as actually take place in the cylinder of the engine is not precisely known.

An approximate form of diagram is therefore taken in theoretical treatment, which diagram is approached more and more closely as the machine is improved. Fig. 50 is such a diagram. AB represents the volume of the mass of steam

when admitted into the cylinder. The *first assumption* is that the pressure of the steam remains constant during admission, so that AB is a line parallel to OX, and the pressure is represented by $OA = GB$. The *second assumption* consists in assigning to the curve BC one or other of two definite forms:

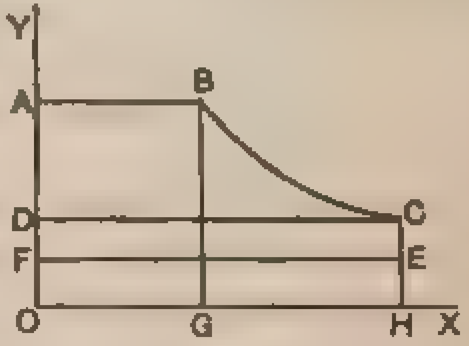

FIG. 50.—IDEAL DIAGRAM.

(I) When the cylinder has no steam-jacket, the steam is assumed to expand without receiving or giving out heat; so that BC is an *adiabatic curve*.

(II) When there is a steam-jacket, it is assumed that the heat communicated by means of that jacket is just sufficient to prevent any appreciable part of the steam from becoming liquid; so that BC is a curve of pressures and volumes of saturated steam.

The real diagram of the ordinary engine is of somewhat different form. The next figure illustrates these differences. The accepted nomenclature is as follows:

The admission line AB is produced by steam on admission. Its normal direction is vertical, or nearly so, as it is traced while the crank is passing its dead-centres. Leaning outward indicates lead. With no lead it would lean inwards, as sometimes with condensing engines.

The steam line BC is traced after the piston has commenced its stroke. Its proper direction is horizontal at a pressure nearly equal to that in the boiler; but this can only be approximated with such openings that the maximum velocity of flow will not exceed about 100 feet per second. But with throttling-engine diagrams the steam line inclines downwards.

The point of cut-off C is the point where the entering steam is cut off. It is usually anticipated by a fall of pressure, which is less as the valve closes more promptly. With some engines, having multiported gridiron valves with detachable valve-gear, this fall of pressure is not appreciable, and the point of cut-off is well defined.

When the instrument has been in good working order, the cut-off may be located at the point of contrary flexure.

The expansion line *CD* begins at cut-off and terminates at exhaust.

The point of exhaust *D* is where exhaust begins; the expansion curve there ends and the pressure begins to fall rapidly.

The exhaust line *DE* is traced while the steam is escaping. When it occupies a considerable part of the return stroke, or nearly all, it indicates a cramped exhaust opening.

The back-pressure line *EF* represents the pressure in the

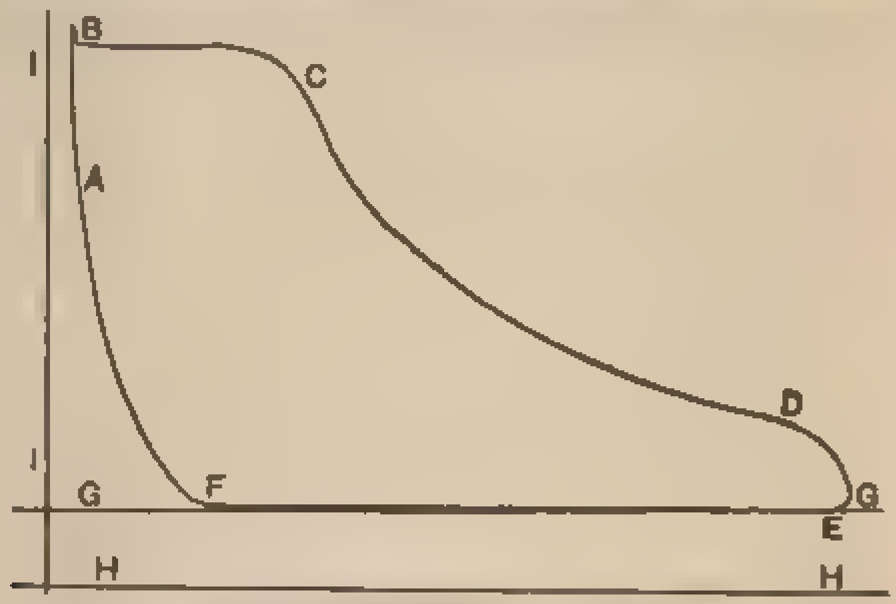

Fig. 51.—Nomenclature of Cards.

cylinder during the return stroke. With non-condensing engines the position of this line is somewhat above atmospheric pressure. With condensing engines it indicates a pressure somewhat in excess of that in the condenser.

The point of exhaust closure *F* is anticipated by a rise of pressure; the eye may locate it very exactly.

The compression curve *FA* exhibits the method and extent of variation of pressure after the exhaust-valve closes.

The atmospheric line *GG* locates the position of equilibrium of the piston of the indicator before steam is introduced.

The vacuum line *HH* is drawn parallel with the atmospheric line at such a distance below it as will measure the pressure of the atmosphere. It is generally placed 14.7 lbs. below the atmospheric line. When a barometer can be con-

sulted, its reading in inches, divided by 2, will give approximately the atmospheric pressure.

The next figure illustrates the form of the diagram obtained from an explosive gas-engine of the Otto type. The three lines, *ABC*, are the result of three successive explosions with varying rates of combustion, *A* indicating rapid, and *C* show-

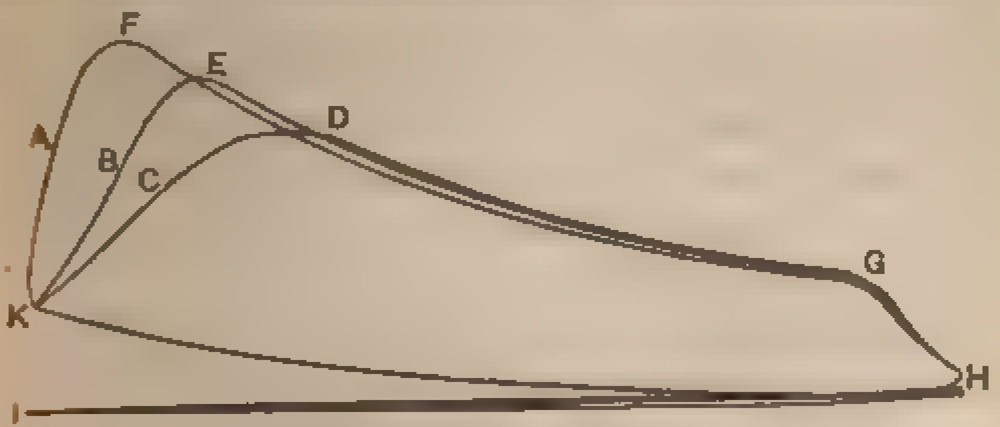

FIG. 52. GAS-ENGINE DIAGRAMS.

ing slow, combustion; neither representing a true explosion, which would have given an initial line above *A*, and vertical. Here the mixture of gases with air enters on the induction-

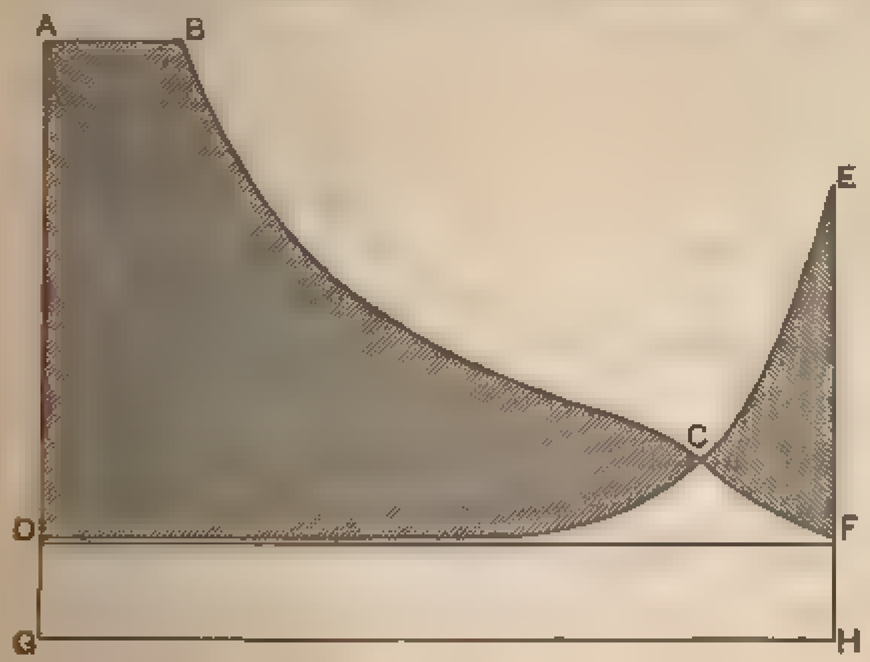

FIG. 53.—EFFORTS ON THE PISTON

stroke *IH*; compression occurs on the return of the piston, *HK*; explosion follows and a second out-stroke, *KFG*; exhaust takes place at *G*; and expulsion of the charge of non-consumed gases takes place on the second return-stroke, *HI*.

The indicator-diagram, although generally assumed to represent the variation of the effort on the piston of the engine at each half-revolution, really exhibits only one part of that action at any given instant. The line *ABCF*, Fig. 53, exhibits the effort of the steam during the forward stroke; but that effort is partly equilibrated by the back-pressure and the compression on the opposite side. If these are represented by the line *DCE*, it is evident that the real variations of net effort are exhibited by the space *ABCD* and by *CEF*; the former being positive, the latter negative. It is thus necessary to combine parts of two opposite simultaneous diagrams to ascertain the real pressures transmitted from the piston.

59. The Causes of Modified Forms of Diagram are usually simple and easily traced. The actual form of the diagram differs from the ideal form, as just described, in consequence of the occurrence of a number of conditions which are usually more or less objectionable. These conditions are classed thus:

Causes which affect the power of the engine, as well as the figure of the diagram:

(1) Wire-drawing in taking steam and at cut-off.
(2) Clearance in the cylinder and passages.
(3) Compression, or cushioning.
(4) Pre-release.
(5) Conduction of heat by the metal of the cylinder.
(6) Liquid water present in the cylinder.

Causes which affect the figure of the diagram only:

(7) Undulations in the motion of the pencil.
(8) Friction of the indicator.
(9) Position of the indicator.

In the accompanying sketch, in which the ideal and a modified form are compared, it is easy to trace some of the causes of difference.

At *A* the pressure of steam is usually a maximum. Should the induction occur at the right time and in the right way, the cylinder will be full of steam at the instant of forward movement of the piston; should the valve open late, *A* will be found

nearer *B*, and the line *KA* will be inclined toward the right; early opening of the induction-port will produce a line starting nearer *M*, and terminating at *A* as at first. If the pressure is not well sustained, *AB* will fall toward *B*; and if the cut-off does not take place promptly, the corner at *B* will be rounded

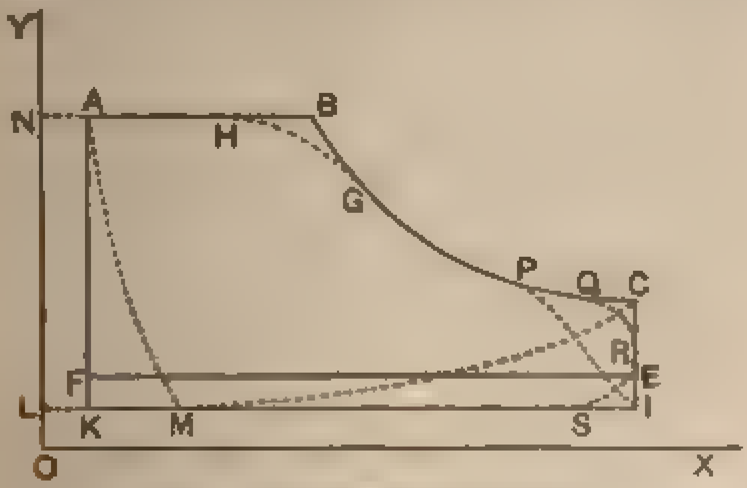

FIG. 54.—DIAGRAMS COMPARED.

off, as from *H* to *G*. At the end *C*, of the expansion-line, similarly, early opening of exhaust will give *QRS* or *PI*; late opening may give *CM*, and the exhaust-line and back-pressure line may become confounded. Early closing of the exhaust-valve may produce a compression-line, *MA*. In all well-designed and properly adjusted steam-engines, this compression, as well as the expansion, will be so arranged as to utilize to best advantage the available heat-energy of the fluid.

Some of these modifications of the ideal diagram are, therefore, due to practical conditions which dictate them. Thus, as the steam-ports are now made, in high-speed engines particularly, it is impossible to secure instantaneously, on opening them, the full pressure of steam in the cylinder; they are therefore given "lead"—opened in advance. The same cause usually retards the inflow of the steam up to the point of cut-off, and thus produces a fall of pressure along the steam-line. Similarly, to meet the disadvantages inherent in the inertia of the fluid, as well as that of practically limited port-area, pre-release of the exhaust-steam is customary. The slower the action of the expansion-valves and of the exhaust-valves, the more are the sharp corners of the ideal diagram rounded off in

the real indicator-card. This action is called "wire-drawing"* the steam. Where the corner at cut-off is obscured in this way, the real point of cut-off may be approximately determined by carrying out the lines *AH* and *PG* to their intersection at *B*, which is taken at the point required. This has been called the point of virtual, or effective, cut-off.*

Wavy lines indicate a defect in the indicator, or its inapplicability at such speeds of engine. They do not always give rise, however, to inaccurate computations. Broken and irregular lines indicate the presence of grit in the instrument.

The accompanying fac-similes of cards taken from a "high-speed" engine well illustrate the method of variation of the diagram, with loads varying from overload to simple friction of engine.

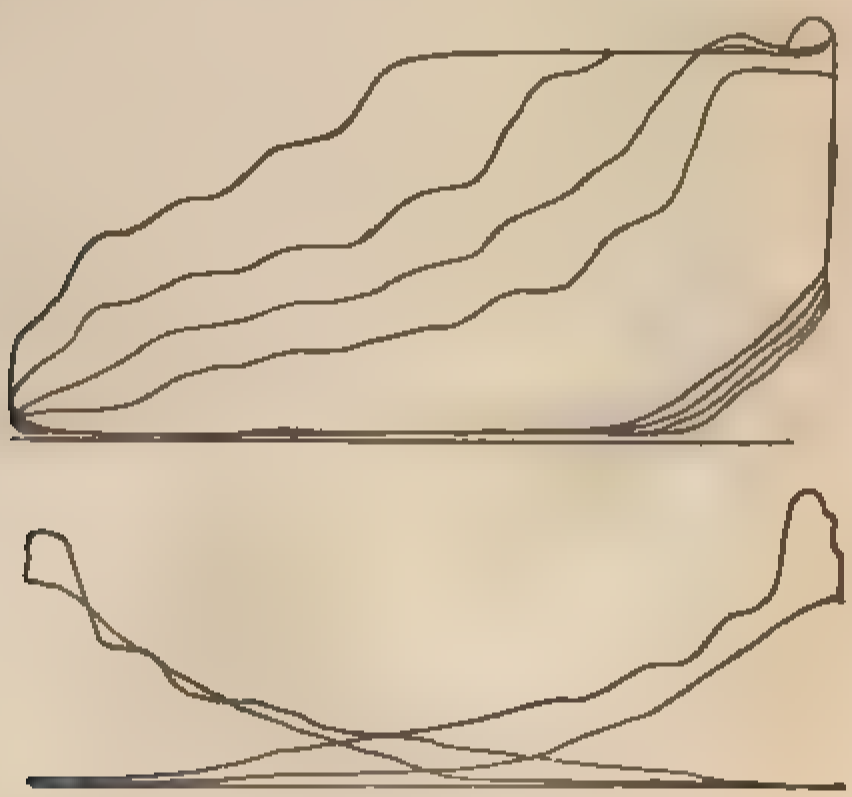

FIG 55.—VARIATIONS OF LOAD.

The data relating to this case are as follow:

Diameter of cylinder . . .	= 8″
Stroke of piston	= 10″
Scale, 60 lbs. to 1 inch.	
Revolutions	= 340 per minute.

* Rankine, Steam engine, p. 418.

Weight of reciprocating parts		= 152 lbs.
Connecting-rod length . . .		= 6 cranks
Maximum valve travel . . .		= $2\frac{5}{8}''$
" lead		= $\frac{1}{8}''$
" port-opening . .		= $\frac{1}{2}''$
Clearance, each end, . . .		= 11%
Maximum, crank end,	M. E. P.	= 61.80 lbs.
	H. P.	= 53.76
" front "	M. E. P.	= 69 lbs.
	H. P.	= 60.03
" mean,	M. E. P.	= 65.40 lbs.
	H. P.	= 56.89
Average initial pressures . .		= 80 lbs.

60. The Interpretation of Diagrams is usually easily effected, and by means of this "engineer's stethoscope" it becomes possible to ascertain the nature and cause of almost every defect in the distribution of pressures and volumes of the working fluid, as in the adjustment of the valve-motion, and the size or proportions of steam-passages, or of the connecting pipes. The power exerted by the steam is easily measurable. These several points may be summarized thus:

(1) Gross power exerted by the steam.
(2) Net power of the steam, and equivalent net power of the engine.
(3) Resistance of unloaded engine.
(4) Net power of the engine.
(5) Details of various wastes of power, as by wire-drawing, back-pressure, etc.
(6) Valve-adjustments.
(7) Effectiveness of valve-gearing.
(8) Adequacy of sizes of port.
(9) Quantity of steam present at any point in the stroke.
(10) Feed-water demanded, exclusive of that wasted by cylinder condensation.
(11) With a boiler-trial, the actual expenditure of steam, fuel, and money, for a given amount of power; and wastes by leakage and condensation.

Of these, the principal are only determined by careful computation, employing as data the quantities graphically measured on the indicator-diagram; others are at once seen by the practised eye, demanding only an inspection of the figures shown on the card. An engine well adapted to its purpose, a perfect engine in the engineer's sense, will usually exhibit an early induction; wide port-opening; an admission-line closely approaching boiler-pressure, and nearly or quite horizontal;

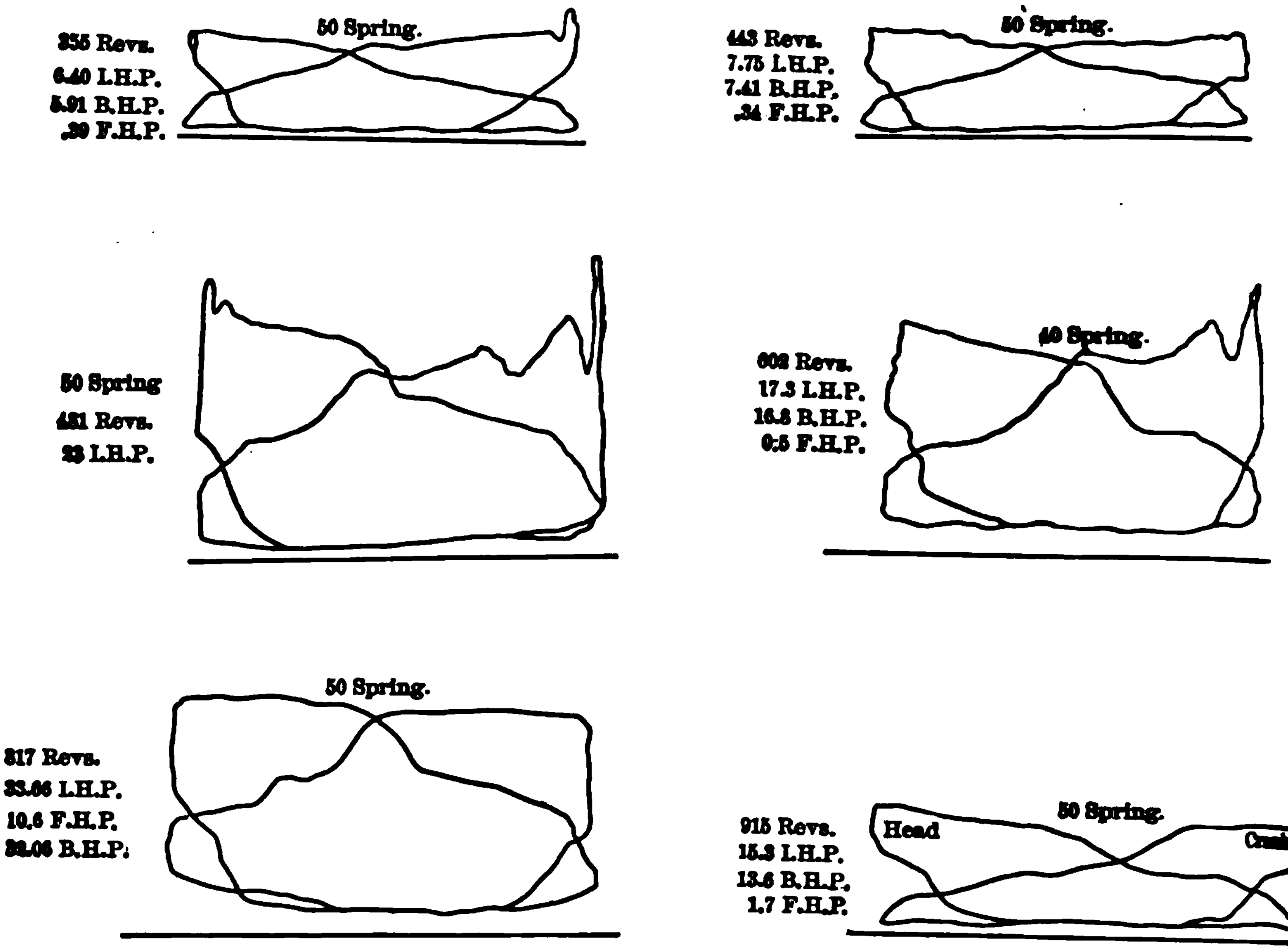

FIG. 56.—EFFECT OF SPEED.

a sharp cut-off; an expansion-line closely approaching the common, or equilateral, hyperbola in form; a somewhat early and a prompt release or exhaust; a low and uniform back-pressure; and a compression carried up well toward initial pressure. These effects are obtained by giving some steam and exhaust lead—greater as speeds and pressures are higher—having good area of ports, securing quick action of the expansion-valve, and a well-adjusted closure of the exhaust-valve. Any depar-

ture from these conditions is ordinarily to be taken as evidence of defective construction or adjustment.

The reduced copies on the opposite page show how the ideal diagram is departed from in the operation of engines, especially at high speeds (Fig. 56).*

All these cards were taken with the same indicators and from the same engine. All exhibit similar departures from the ideal form of diagram, and all illustrate well the two kinds of effect already described—those due to practical conditions of construction and operation of the engine, and those produced by the inertia and friction of the indicator.

As an illustration of the interpretation of the diagram, we may take the following example (Fig. 57): †

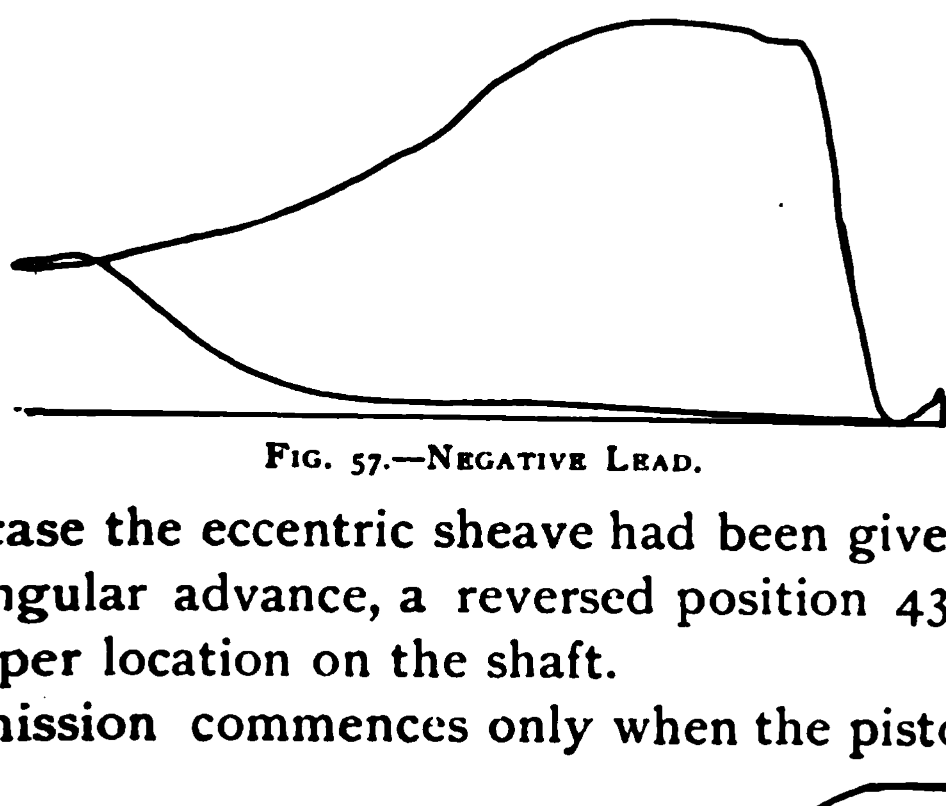

FIG. 57.—NEGATIVE LEAD.

In this case the eccentric sheave had been given, instead of the usual angular advance, a reversed position 43 degrees behind its proper location on the shaft.

The admission commences only when the piston has trav-

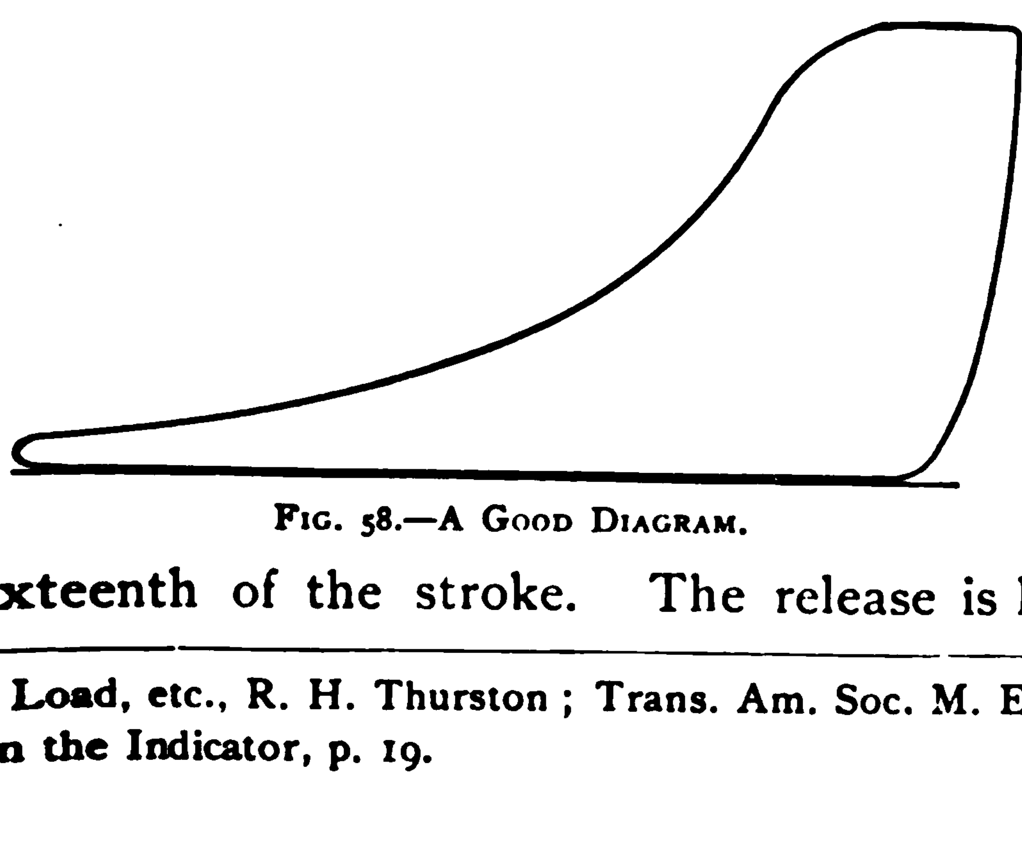

FIG. 58.—A GOOD DIAGRAM.

elled one-sixteenth of the stroke. The release is late by an

* Variable Load, etc., R. H. Thurston; Trans. Am. Soc. M. E., 1888.
† Barrus on the Indicator, p. 19.

average similar amount. The pressure before cut-off is low, the back-pressure high, and there is no compression.

Waste of power is here evidently produced; the expansion is too early terminated and the exhaust is deferred, wasting steam in even higher degree than power of engine. The succeeding figure represents as close an approach to the ideal form as is often seen, and probably has too high a ratio of expansion to give best results.

The accompanying diagram, taken by Mr. King * from the condensing engines of the "Powhatan," and the dotted variations from the actual line, exhibit again the various principal deductions to be made.

Fig. 59 is what would be termed a good diagram.

Steam........	10	"Powhatan" stb. cylinder, bottom.
Vacuum.......	27	Nov 7, 1855. 10 A M.
Hot-well......	106 Fahr.	One engine and one wheel in
Revolutions.....	9.5	operation.
Throttle........	8.	Smooth sea.

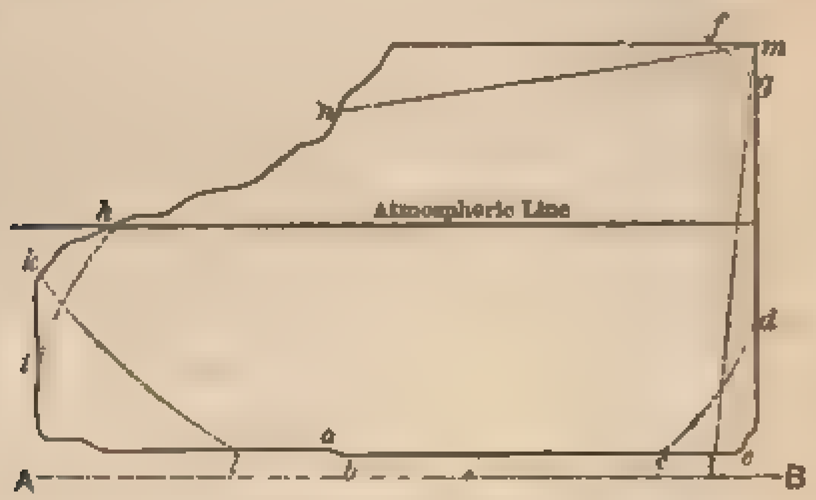

FIG. 59.—TYPICAL DIAGRAM.

It appears, however, that the piston of the indicator worked tightly, which occasioned it to stick in places, as is evidenced by the steps in the expansion-line, and also at *ab* in the vacuum-line.

Should Fig. 59, instead of as shown, have the lower right-hand corner cut off as at *cd*, the exhaust-valve closed too soon,—at *c* instead of *e*, occasioning cushioning.

* Practical Notes, p. 46.

Had the upper right-hand corner been as shown by the line *fg*, the steam-valve must have opened too late. Had the exhaust corner been cut-off, as shown by *hi*, the exhaust-valve would have opened too soon; but had it been at *kl*, it would have opened too late, and would move too slowly, preventing free escape of steam; or the exhaust-passages would have been too small, which would produce a similar effect. Had the steam-line fallen as at *mn*, it would have shown that the throttle was partially closed, or the steam-passages too small. Should there be excessive lead to the steam-valve, the line *dm* will have the top inclined to the right as from *L* to *m*. Late opening would produce an inclination in the opposite direction.

FIG. 60.—UNSYMMETRICAL VALVE-SETTING.

This figure is a double diagram taken from one of the paddle engines of the "Great Eastern," when on her trial-trip in the British Channel.

It will be observed that the valves were unevenly set. The diagram from the top of the cylinder shows that the pressure on the piston was 20 pounds, cut-off at one-third the length of the stroke, and expanded down to atmospheric pressure at the termination. The diagram from the bottom of the cylinder shows that the steam was at 22 pounds, cut-off at half-stroke, and expanding to 4 pounds above the atmosphere at the termination; in both cases the vacuum being 12 pounds or about 24 inches. The number of strokes was $11\frac{3}{4}$, and the speed of the piston 331 feet per minute. The exhaust closed when the piston had travelled five-sixths of the length of the cylinder,

checking the progress of the piston when about two feet from the end of the cylinder.

Whenever the adjustment of valves is proposed, guided by the indicator, it should be carefully noted whether parts affected by such adjustment are liable to injury by the change. Slide-valves, for example, which have been long at work, sometimes wear their seats to a shoulder when they are not terminated by a depression which is overrun by the valve. If this shoulder is not first removed, the change may cause leakage, or even accident.

61. Compound-engine Diagrams, as produced by the steam-engine indicator, often differ even more than with the simple engine, from that ideal "card" which would be given were the expansion precisely as intended, and the engine free from defect, as in clearance-spaces and port-resistances. The best compound engines show considerable loss, as has been seen, in these ways, and also in that drop of pressure between high- and low-pressure cylinders which often constitutes a very sensible source of waste of heat, steam, and fuel. Where the Wolff system is adopted, however, if the load be constant and the machine well proportioned to its work, and if the "dead-spaces" can be made small, the approximation of the actual to the ideal card may be very close, as is illustrated by the accompanying pair of diagrams from a pumping-engine of this character.

The action of the steam and its variations of pressure are here seen throughout the cycle to be precisely similar to that in a simple engine. Large steam-ports and a good expansion-gear bring the steam-line close up to that of boiler-pressure; a well-jacketed cylinder allows the expansion-line to follow closely that laid down for the ideal engine, short and free ports between the two cylinders give an exhaust from the high-pressure and a supply to the low-pressure cylinder which are nearly coincident; and the two cards would, if reduced to a single diagram, exhibit a very close approximation to that which would have been constructed as the ideal diagram of this class of engine. Such satisfactory results are rare; and in most

cases the differences between the actual and the ideal case are very marked, and are serious in their effect upon efficiency.

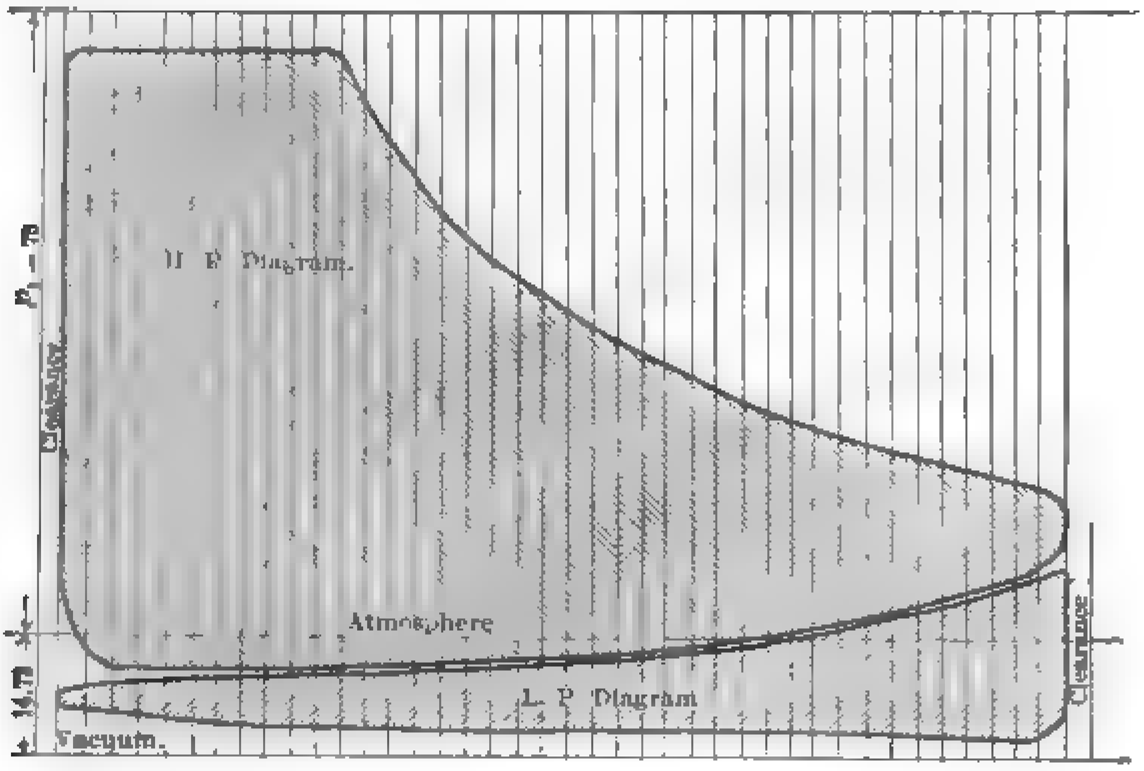

FIG. 61.—ACTION OF STEAM; WOLFF ENGINE.

The next diagrams represent the pair of cards taken from a well-known single-acting compound engine of small size. The

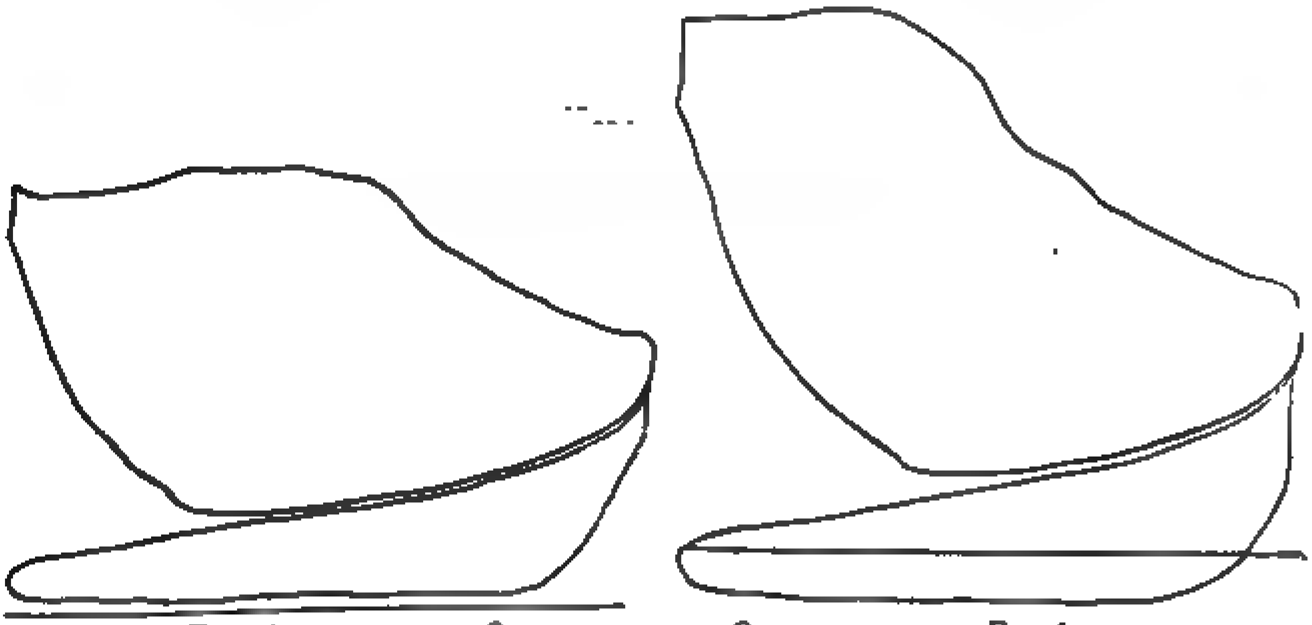

FIG. 62. COMPOUND-ENGINE CARDS. FIG. 63.

action is precisely as before, except that heavy compression is introduced to fill the comparatively large clearance-spaces.

This distribution is peculiarly well adapted to high-speed practice, such as it actually illustrates. The small loss of pressure between the two cylinders well exhibits this special advantage of well-proportioned engines of the Wolff type.

The Construction of Diagrams exhibiting the method of action of steam in the cylinders of the compound engine, as a preliminary to the settlement of the details of the design, is usually as below; these ideal or theoretical indicator-cards indicating the ideal action of the type of engine proposed, as modified by such conditions of operation as the designer can with more or less exactness define and represent. The separate diagrams appertaining to the two cylinders, high- and low-pressure, of an ideal compound engine, are to be combined in a manner indicated as follows,—to obtain a single diagram representing the complete cycle of changes of pressure and volume of the steam from the moment of entrance up to that of its discharge,—the Wolff type of engine being chosen for illustration: The steam being admitted into the smaller cylinder until it fills a volume, represented by $\overline{BC}$ in Fig. 64, the absolute pressure is represented by BO above zero on POQ. The steam is then cut off, and it expands with a pressure gradually diminishing, as shown by the curve CD. DN being perpendicular to OQ, $\overline{ON}$ represents the space swept through by the piston of the small cylinder. Next, a communication is opened between the small and the large cylinder; and the forward stroke of the large piston takes place at the same time with the return stroke of the small cylinder. Thus the steam is driven before the piston of the small cylinder, and drives that of the larger cylinder, and it exerts more energy on the latter than it receives from the former, as the piston of the large cylinder sweeps through greater space, the difference between those quantities of energy is added to the energy exerted on the piston of the small cylinder. This

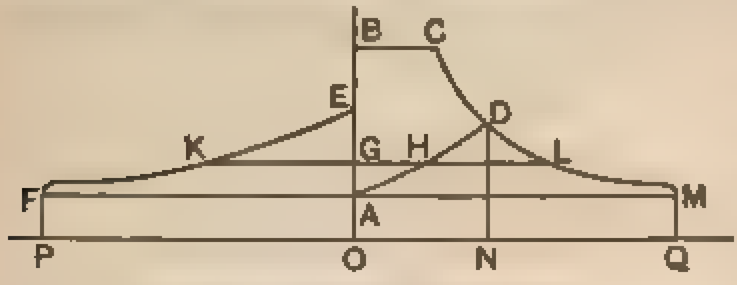

FIG 64.—COMBINED DIAGRAMS.

action is represented by DA and EF: the ordinates of DA representing the backward pressures in the small cylinder, and EF the forward pressures in the large cylinder.

During return stroke of the larger cylinder the steam is expelled, exerting a back-pressure along FA, while steam is admitted again into the small cylinder, and expanded during a new stroke of that cylinder.

Thus are obtained the diagrams $BCDAB$ for the small cylinder and $EFAE$ for the large, and the sum of their areas represents the energy exerted by the quantity of steam expended.

When the diagrams are to be used for the purpose of studying the relations between heat expended and work performed, it is best to combine them into one diagram, thus:

Draw a line KGH parallel to POQ, intersecting both diagrams, and lay off upon it $HL = KG$; and $GL = GH + KG$ represents the total volume in both of the steam-cylinders, when its pressure is OG; while L is a point which would have been reached had the action taken place in the large cylinder alone.

By drawing a number of lines, as KL, any number of points may be found to complete the *combined diagram* $BCDLMAB$, whose length $OQ = OP$ represents the volume of the large cylinder; and this diagram may be discussed as if it represented the action of the steam in the large cylinder only.

Thus, as observed by Rankine,* *the energy exerted by a given portion of a fluid during a given series of changes of pressure and volume depends on that series of changes, and not on the number and arrangement of the cylinders in which those changes are undergone.*

When the diagrams taken from the two cylinders of the simplest form of compound engine are placed together, having the same length, they form a whole such as is illustrated in Figs. 61, 62, and 63. Since, however, the diagram should be constructed so as to make the horizontal scale one of volumes, the pair cannot be compared with the card taken from a simple engine, and they must be reduced to a common scale of vol-

* Steam-engine, § 261, p. 336.

umes by either reconstructing the smaller on the volume scale of the larger, or *vice versa*. The latter is the usual course, as is illustrated in the succeeding examples of compound-engine diagrams. In thus combining these diagrams, it must be remembered that the work represented is that of the engine for one stroke or is that of a single charge of steam, just as in the single cylinder. The base-line measures the stroke, or the volume swept through by the piston in one forward movement. To make a combined diagram, each abscissa of the low-pressure diagram must be increased in such a proportion as to cause it to become proportional to the total volume assumed by the steam at the instant of the production of that line. The ratio of the enlargement is that of the effective capacities of the two cylinders. If the diagrams have both been taken with the same indicator-spring, the two diagrams may now be adjusted to make a single one which exhibits, at all periods of its cycle, the actual relations of pressure and volume of the steam. Were the engine perfect in its proportions and its operation, the two figures would produce a combined diagram precisely like that which might be obtained from a single engine working the same weight and pressure of steam with the same expansion-ratio.

At any point in the simultaneous motion of the two pistons the volume of the steam will evidently be the sum of the volumes, v_x traversed by the large, and V_x still to be moved through by the small, piston; and if the part of the volume traversed in each cylinder is x, this total volume, as to x, is

$$v - x + \frac{V}{v}x = v_x + V_x ;$$

and the pressure at this point is measured by the ratio of the volumes; so that

$$p_x = \frac{p_1 v \div r_1}{v_x + V_x} = \frac{p_1 v}{r_1(v_x + V_x)} = \frac{p_1 v}{r_1 v + (r - r_1)x}.$$

This value is somewhat modified by the presence of the intermediate passages between the cylinders, a drop occurring in the pressure at the instant of opening the exhaust from the small cylinder; but this drop is less as those passages are larger; and if forming an intermediate reservoir, as is sometimes the case where "reheating" between the cylinders is practised, this loss and the corresponding reduction in the mean pressure obtained, in work done, and in the actual total ratio of expansion, is sometimes quite unimportant compared with the gain by that process. A common value for the reduction of total expansion is not far from twenty per cent., rising to one-third with small reservoirs and falling to a lower figure with larger spaces. The loss of work may usually be neglected.

The receiver type of engine with equidistant cranks and intermediate reservoirs is less seriously affected by intermediate spaces. The reduction of pressure and the loss of total expansion is but about ten per cent., where the receiver-space is equal to the volume of the smaller cylinder, and falls to less than five, in usual cases, when the receiver is as large as the larger cylinder; losses which may be easily approximately estimated and allowed for in any case.*

In the next illustration from Mr. Porter's report, the natural form of the expansion-line, in the single cylinder, having the capacity here observed in the low-pressure engine, would be that shown by one or other of the two dotted lines, accordingly as the expansion approached more or less closely the hyperbolic form. The initial volume is AB, and the pressure as shown on the vertical scale; while the gradual loss of pressure with increase of volume is shown by the two scales as the line progresses toward the right to its terminal point at I. The deviation from the dotted line of the actual expansion-line between B and C illustrates the gain of weight and pressure due to the progressing re-evaporation of steam originally condensed in the cylinder at the opening of the steam-valve, and to the admission of the fluid into the colder cylinder. Here

* For exact expressions, see Sennett, Appendix; and Clarke's Manual pp. 849 *et seq.*

expansion occurs from the initial pressure and volume at *B* down to the terminal point *C* in the high-pressure, and from *C* or *H* to *I* in the low pressure, cylinder. The indicator-diagrams actually obtained are *ABCD* and *EFG*, the latter being the equivalent in the low-pressure cylinder of the card *HIJ*,

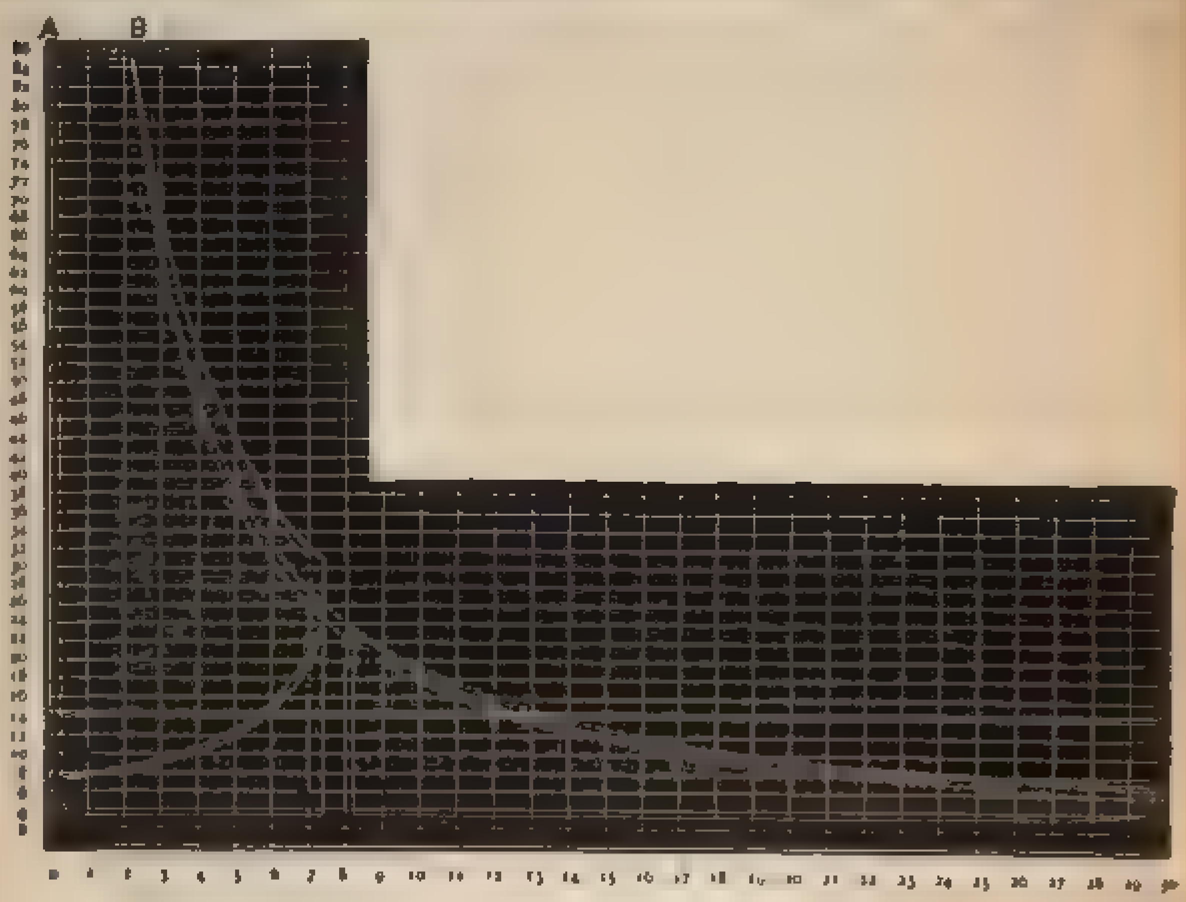

FIG. 65.—COMPOUND-ENGINE DIAGRAMS.

which would have been produced had the high-pressure cylinder been given sufficient length to permit the completion of the expansion in that cylinder. The variation of the full line, representing the real diagram, from the ideal dotted expansion-line is indicative of the fluctuations of pressure produced by the condensation and re-evaporations taking place as expansion progresses in the metallic chamber serving as working cylinder.

The succeeding figure illustrates the visible differences between the diagrams actually taken from the two cylinders of a compound engine—in this case a "Reynolds-Corliss"—and the ideal combined card.

This next diagram, from an engine of similar class with the

preceding and published by its designer, exhibits at once the method of reducing the actual indicator-diagrams to the combined form, and the variations from the ideal expansion-line due to imperfections of the engine as a work of human art. Pressures are measured in pounds on the square inch and volumes in cubic feet, actual capacities of cylinder being given. As shown on the diagram, about $3\frac{1}{8}$ cubic feet of steam enter

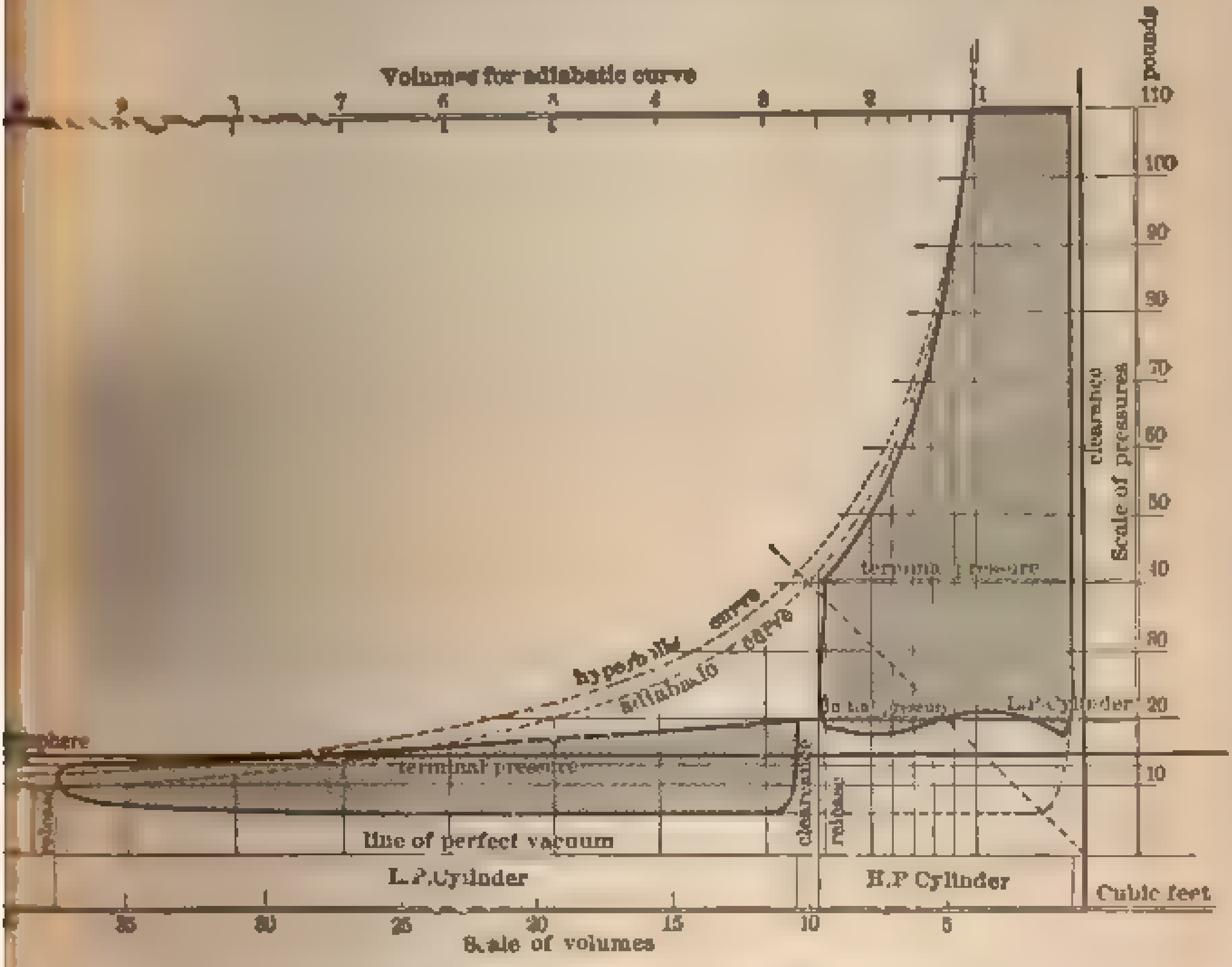

FIG. 66.—ACTUAL AND IDEAL DIAGRAMS, COMPOUND ENGINE.

the high-pressure cylinder, each stroke, at a pressure of 110 pounds per square inch above vacuum: it expands nearly adiabatically to $9\frac{1}{4}$ cubic feet, is then transferred to the low-pressure, dropping from the terminal pressure, 40 pounds in the high-pressure cylinder to 20 in the low-pressure, and then expanding in the latter down to about 12 pounds when it passes into the condenser, the back-pressure thus becoming not far from an average of 6 pounds. The two indicator-diagrams

are shown by the "hatched" spaces; the ideal diagram incloses both, its outline being the dotted lines. The very considerable space measuring the difference of the two areas is a gauge of the imperfection of the cycle. The departure of the actual line from the two ideal expansion curves, and the fact that the former lies within both the latter, indicate that the jacket does not supply heat enough to compensate the condensation of the expanding fluid; far less enough to retain its temperature constant or to continuously superheat it.

The discordant fluctuations of similar lines in the two indicator-diagrams exhibit the effect of non-synchronous motion of the two cylinders.

The accompanying illustration exhibits the proportions of the diagrams taken from a triple-expansion engine, drawn to common volume and pressure scales, and placed under the Mariotte line. The engine has cylinders having the ratios

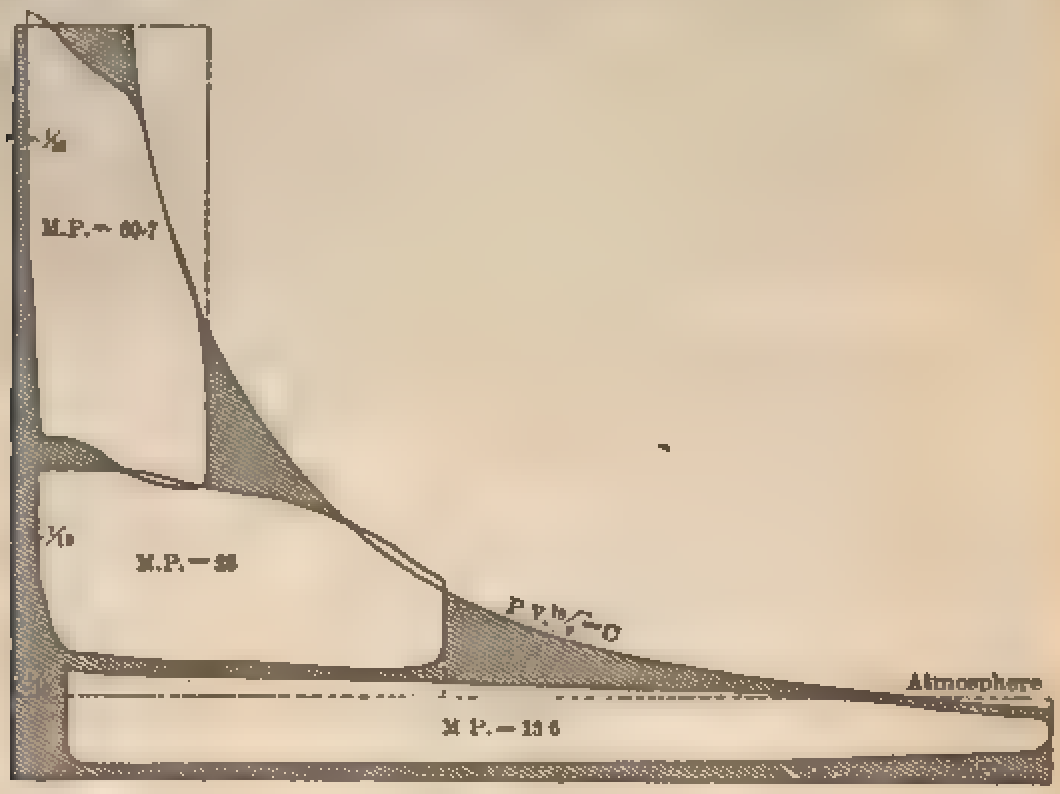

FIG. 67.—TRIPLE EXPANSION DIAGRAM.

1 : 2.25 : 2.42, and the total ratio of expansion is 8, the cut-off in the several cylinders being set at 1.47, 1.3, 1.3. An advantage of this type of receiver engine, with its cranks making equal angles, is that the drop in pressure may be reduced to an

unimportant amount. Here steam-pressure is carried at 125 pounds by gauge; the efficiency 0.18, and the coal used 1.5 pounds per I. H. P. per hour.

62. Special Applications of the indicator are of peculiar interest to the engineer. Valve-adjustment is often performed, and should always be checked, by the aid of this instrument. The application of the indicator to the steam chest, and the comparison of its readings with those of the ordinary diagrams and of the steam-gauge at the boiler, will often reveal defects in the steam-passages or valve-action otherwise difficult of detection; its use on the air-pumps of condensing engines and on the main pumps of pumping-engines similarly reveals anything objectionable in their construction and operation; and the motion being derived from the eccentric or the valve-mechanism when attached to the engine in the usual manner will permit a more minute examination of those phases of operation which are not easily studied on the common form of card. In some cases a continuous motion, derived from the crank-shaft, is adopted for this purpose.

In valve-adjustment, an inspection of the diagram shows the operation of the valve-mechanism as set. The necessity of adding lead or the reverse, of resetting the valves and eccentrics, is seen, and they are readjusted and diagrams again taken, these operations being repeated until the form of diagram desired is approximated as closely as is practicable.

In illustration of the forms of diagram obtained from the steam-chest and their interpretation, Mr. Porter gives those exhibited in the illustration.*

In these figures the steam-pressure fluctuates in the valve-chest with the draught upon it by the engine, rising to boiler-pressure after cut-off takes place, and falling below it more or less as the steam enters the valve-ports more or less rapidly. The extremities of the diagram correspond to the end of stroke of engine-piston, the points *c* to the points of cut-off at each stroke. The lower line shows the pressure in the chest during the interval up to the moment of action of the expansion-valve.

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* The Richards Steam-engine Indicator, pp. 177–8.

Immediately afterward the pressure rises to boiler-pressure and there remains until the point is reached at which a drop of the upper line shows that the opposite end of the cylinder has begun to take steam. In *A* and *B* the pressure, when at its maximum, actually exceeds boiler-pressure, the surge of the mass in the steam-pipes and chest, when suddenly checked, causing a wave of abnormally high pressure. These were taken from Mr. Porter's engine at the Paris Exposition of 1867, when making 200 revolutions per minute.

In *C*, evidence is found of insufficient area of steam-pipe,

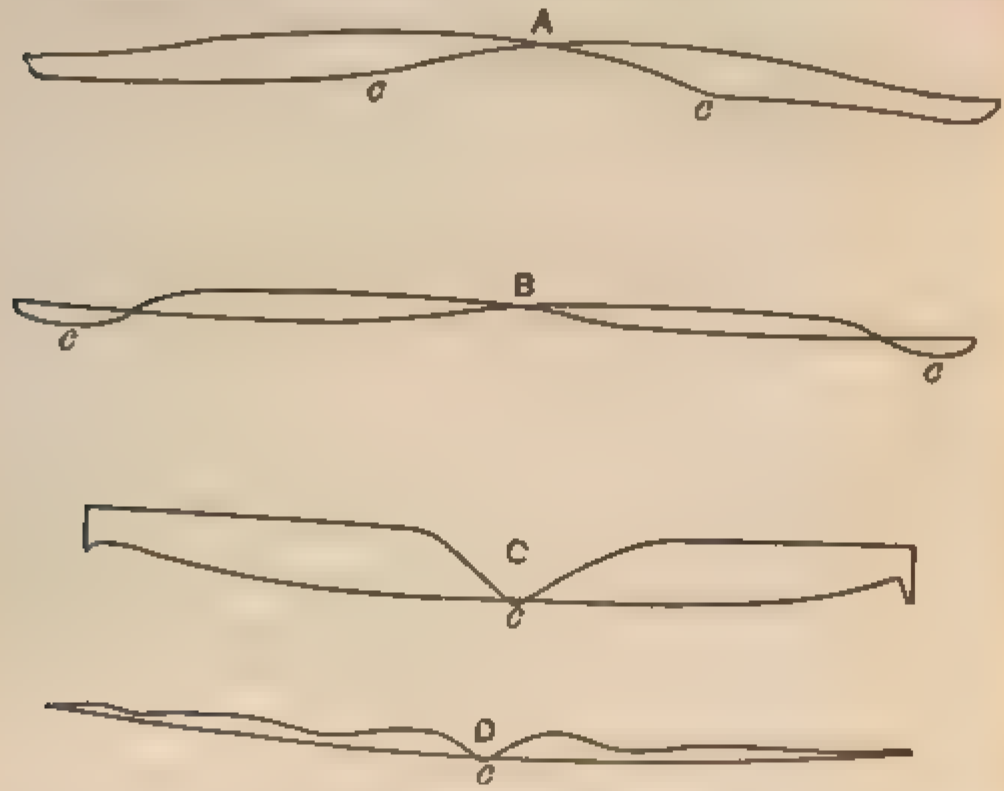

FIG. 68.—VALVE-CHEST DIAGRAMS.

resulting in the observed considerable fall of pressure when the engine takes steam at either extremity and the correspondingly large rise at the cut-off, *c*. The pipe having been enlarged, the card *D* was obtained; the sudden drop and continuous fall of pressure while the engine takes steam and the considerable loss of pressure, of power, and of efficiency indicated in *C* are in *D* all absent.

63. **Pump Diagrams** obtained by application of the instrument to the air-pump of a condensing engine are seen illustrated in the two succeeding figures, given by Mr. King, as

taken by him from the air-pumps of the U. S. S. "Powhatan," a paddle steamer of old type, having jet-condensers.*

With the first of the pair the engine was working as usual; in the second case the pump was taking in a large excess of air through the bilge-injection. The pump, in the first ex-

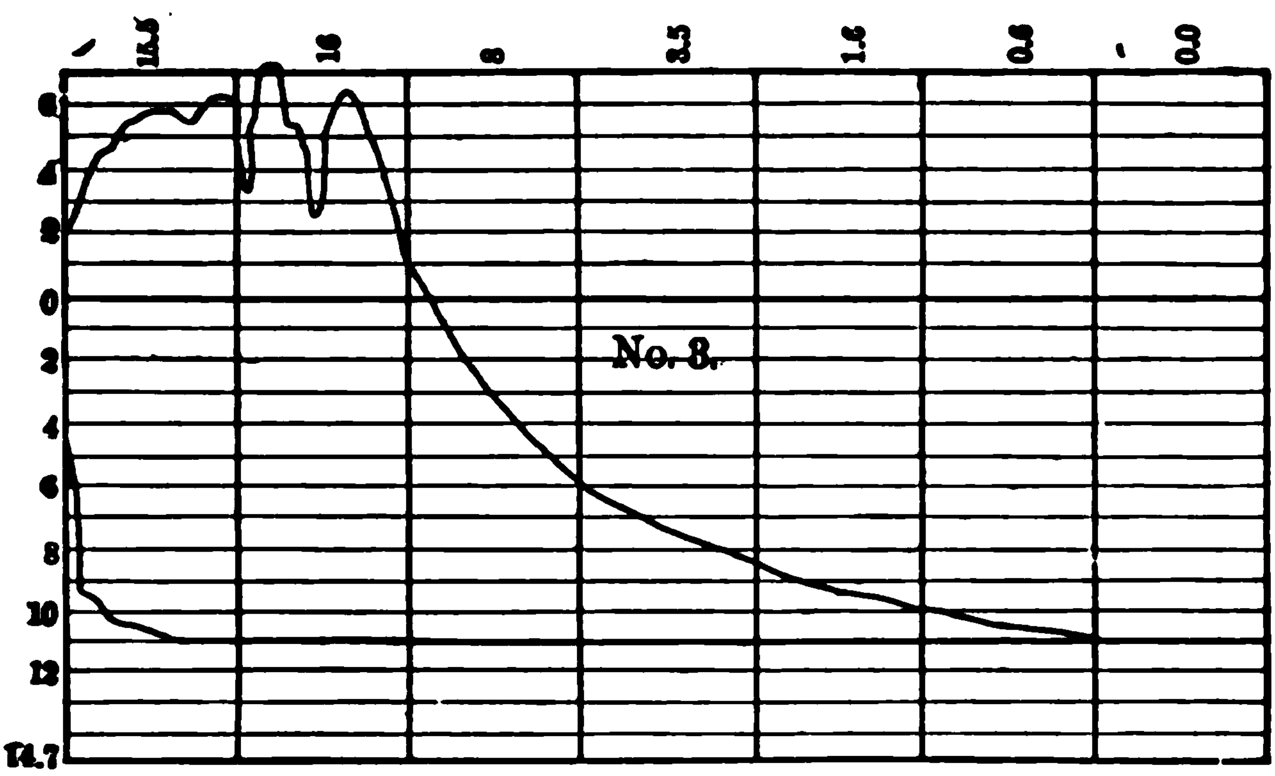

FIG. 69.—NORMAL AIR-PUMP CARD.

ample, draws the mingled air, vapor, and water from the condenser at a pressure about 4 pounds above a vacuum, throughout the induction-stroke, and on the return or educ-

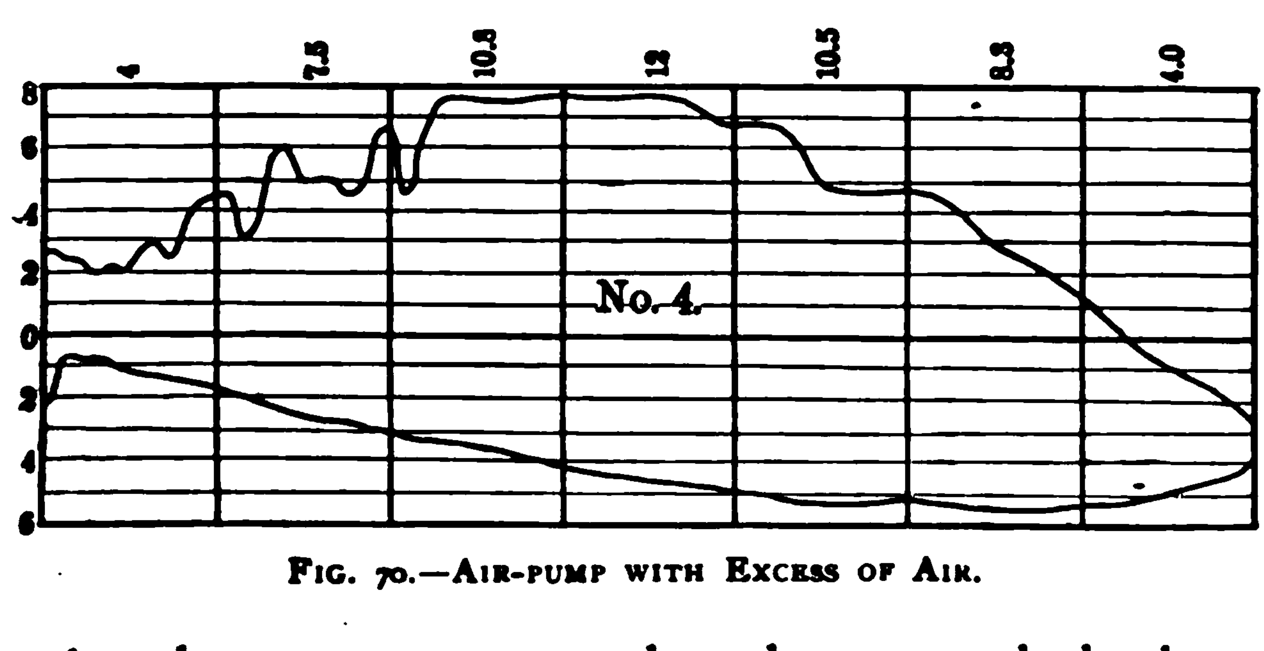

FIG. 70.—AIR-PUMP WITH EXCESS OF AIR.

tion-stroke they are compressed under a regularly increasing pressure to the point at which, the delivery-valve opening about 5 pounds above atmospheric pressure, the whole mass is discharged with fluctuations of the line due to rise and fall of the

* Practical Notes, pp. 56–60.

valve and irregular expulsion of air and vapor. The pressure then falls to about 2 pounds, and the end of the stroke is reached. Excess above the last observed pressure is due to friction in the discharge-passages and delivery-pipes. These diagrams are seen to be characteristically different from those obtained from the engine.

Pumps raising water, or any other incompressible fluid, should give a diagram like that here shown, as taken from the remarkable pumping-engine built by Mr. Corliss for the town of Pawtucket, R. I. This diagram is seen to be perfectly

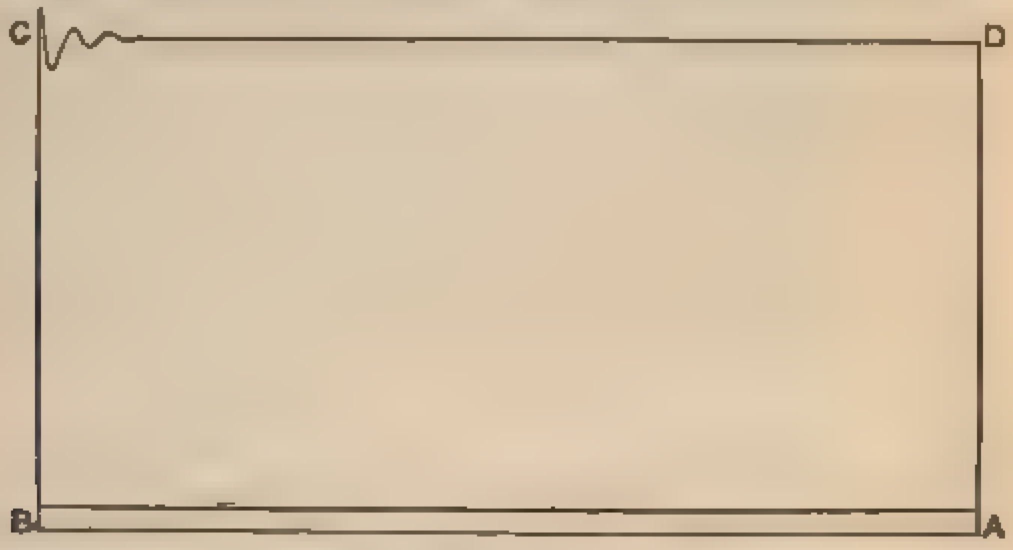

FIG. 71.—PUMP DIAGRAM.

rectangular, the water entering from beginning to end of stroke *AB*, the pressure about five pounds below that of the atmosphere; the change to the delivery pressure, *BC*, a little above 100 pounds per square inch, taking place instantly on reversal, and the discharge, *CD*, occurring at uniform pressure. The slight disturbance at one corner, *C*, is due to a jar of the spring of the instrument.

Air-compression Diagrams exhibit this effect, as in the adjacent figures, showing a steam-engine and an air-compressor card taken from the Allen "positive valve-motion" compressor at the same time. The engine drives the compressor, and the work shown on the engine-card exceeds that of the compressor by the lost work of the apparatus. The engine is seen to have early admission, incomplete expansion, early release, and cushioning, all of which are practically to some extent required for

best effect; but the compressor takes in its charge throughout the induction-stroke, compresses steadily from the minimum to the maximum pressure, and has no observable variations from the ideal diagram such as are exhibited in the case of the steam-engine. This machine was running at 144 revolutions

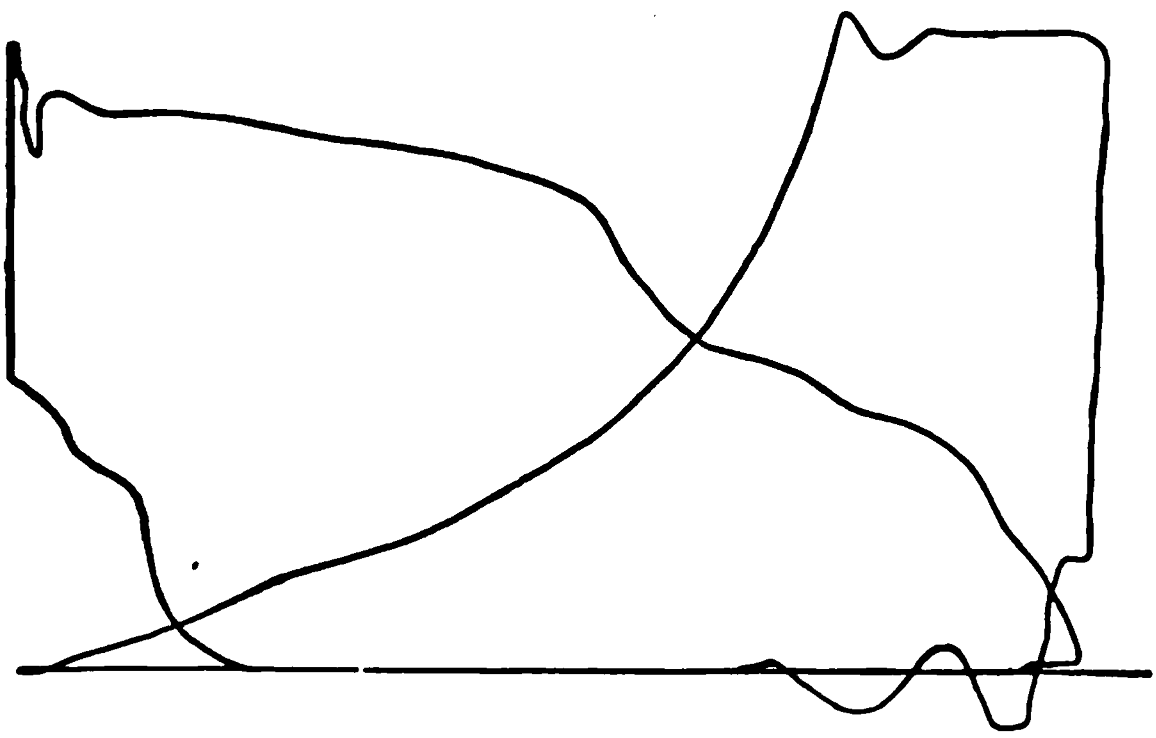

FIG. 72.—STEAM-ENGINE AND AIR-COMPRESSOR DIAGRAMS.

per minute when "indicated," and this speed caused those fluctuations of the curve due to inertia in the instrument.

Diagrams taken with the motion of the paper derived from the main shaft are of the form seen in the next figure.

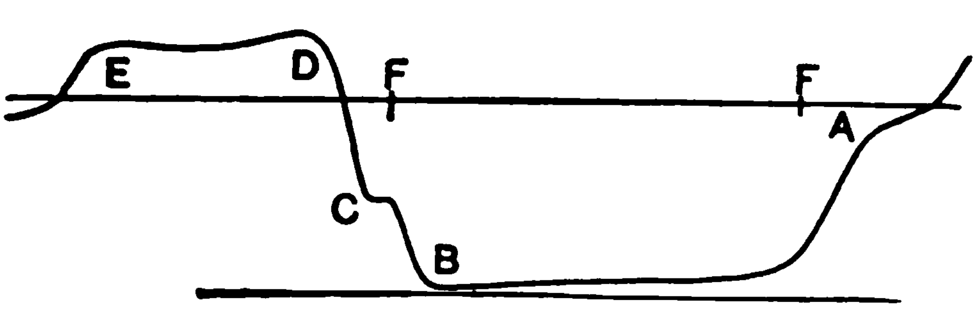

FIG. 73.—SHAFT DIAGRAM.

From *A* to *B* is the exhaust and vacuum line; from *B* to *C*, the cushioning; *C* to *D*, receiving-line; *D* to *E*, steam-line; *E* to *A*, expansion; *FF* are supposed to be the ends of the stroke.

Fig. 74 is the same diagram extended more nearly to its original length, the ends joined and then folded at *FF* so as to represent more nearly the usual diagram, but still preserving the peculiar length and proportion of lines. Diagrams taken in this manner expose more perfectly defective arrangements of valves; and a studious comparison of them in connection with

the usual form may often be useful in detecting defective sizes and proportions of valves, steam-passages, etc., and show very

Fig. 74.—Reconstructed Card.

correctly the proportion of time occupied in each different operation during the revolution of the engine.*

Attaching the line to the eccentric motion will give similarly useful diagrams.

64. Peculiar Forms of diagram are often met with, each invariably exemplifying some singular or abnormal arrangement of the engine.

Steam in boilers....	9 lbs.	"Powhatan," Feb. 13, 1854.
Revolutions........	5	Stb. cylinder bottom, working by hand.
Hot-well...........	100°	
Throttle...........	4	

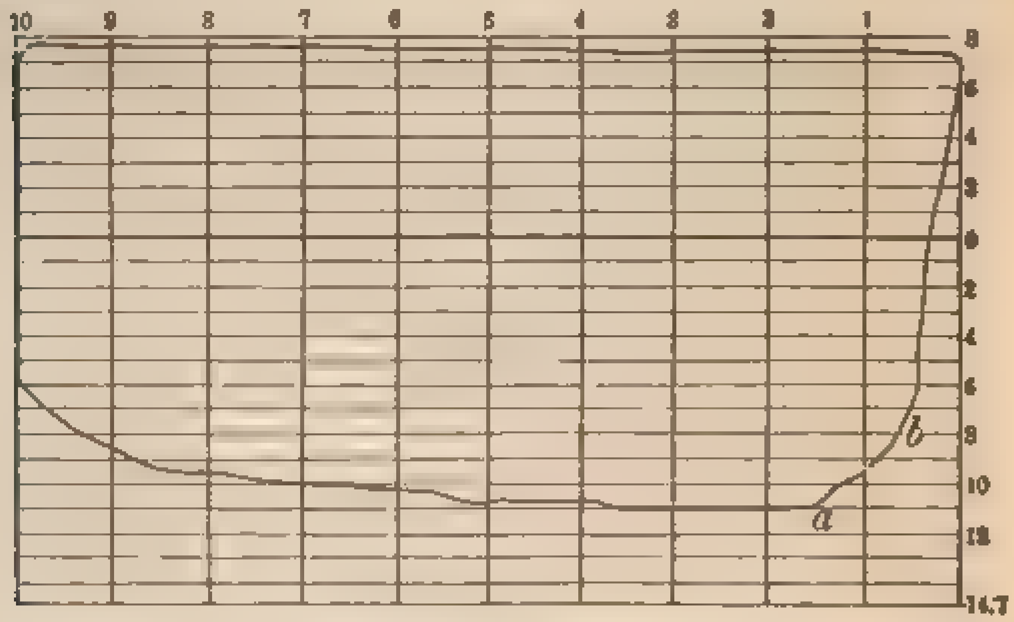

Fig. 75.—Working by Hand.

Fig. 75, from King, is a diagram showing the operation of the valves while working by hand. This valve exhibits large

* Stillman on the Indicator, p. 31.

cushioning and steam lead, the exhaust-valve closing at *a*, and the steam-valve opening at *b*, so that the engine passed the centre against a pressure of 6¼ pounds above the atmosphere.

Fig. 76 is obtained from the same source as the last. In this case the engine was working as a non-condensing engine with a

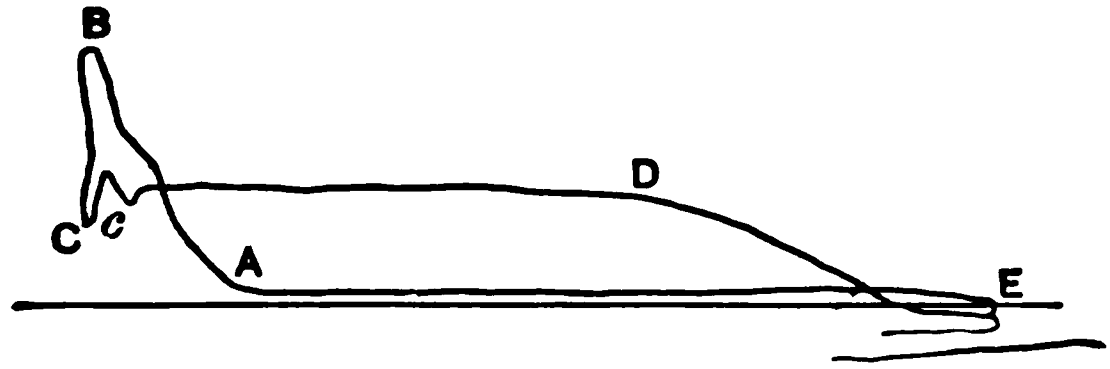

FIG. 76.—PECULIAR ADJUSTMENTS.

very low pressure of steam. The exhaust closes at *A* causing the pent-up steam to be compressed to *B*, where the steam-valve opens, and the pressure in the cylinder, being greater than that in the boiler, immediately falls to *C*. The hook at *C* is occasioned by the momentum of the indicator-piston. At *D* the cut-off closes, causing the steam to be expanded to *E*, below the atmosphere. At *E* the exhaust-valve opens and the pressure rises up equal to the back-pressure, causing the loop on that corner of the diagram.

Fig. 77 is a diagram from a non-condensing engine. The

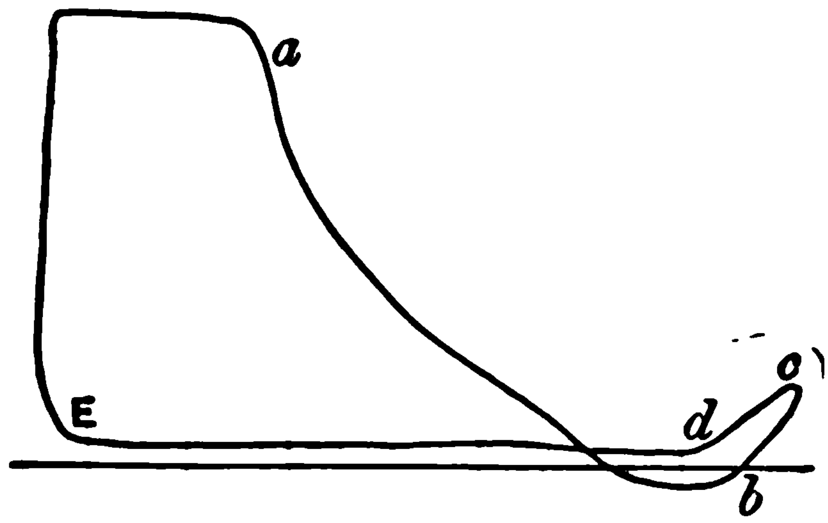

FIG. 77.—READMISSION.

pressure rises from *b* to *c*; but supposing the exhaust to open at *b*, there could be no reason why the pressure should rise beyond *d*, the amount of back-pressure on the opposite side of the piston; such a diagram could only be formed from a slide-valve engine, in this manner: Steam being admitted until the

piston arrived at *a*, the independent slide-valve cut off the steam; it was then expanded to the point *b*; at *b*—the steam-valve having deficient lap and lead, and thus being open—the cut-off valve opened, admitting fresh steam, which caused the line *bc* to be traced. At *c* the steam is shut off by the steam-valve itself and the exhaust opened; the pressure then falls to *d*, and the exhaust-line is traced.

Fig. 78 is an Otto gas-engine diagram, taken purposely, by

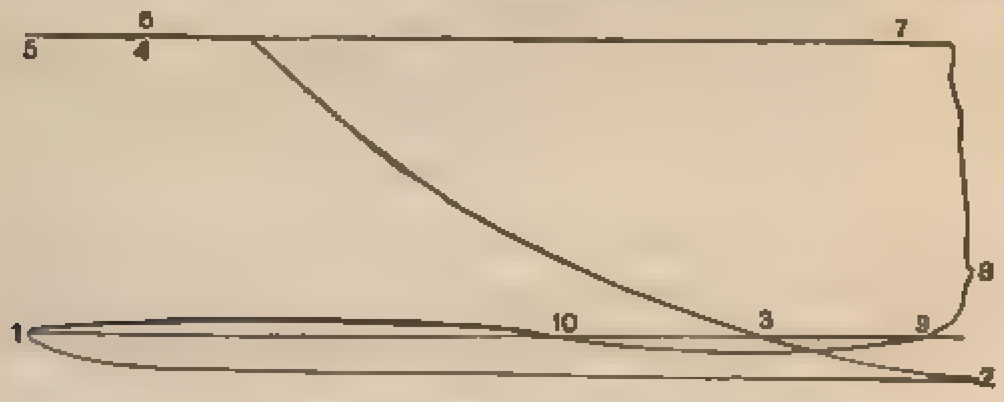

FIG. 78. DIAGRAM TAKEN WITH LIGHT SPRING.

Messrs. Brooks and Stewart, with a light spring, in order to exhibit better the action of the machine at the more obscure points.* The induction-stroke begins at 1; the mixed gases are

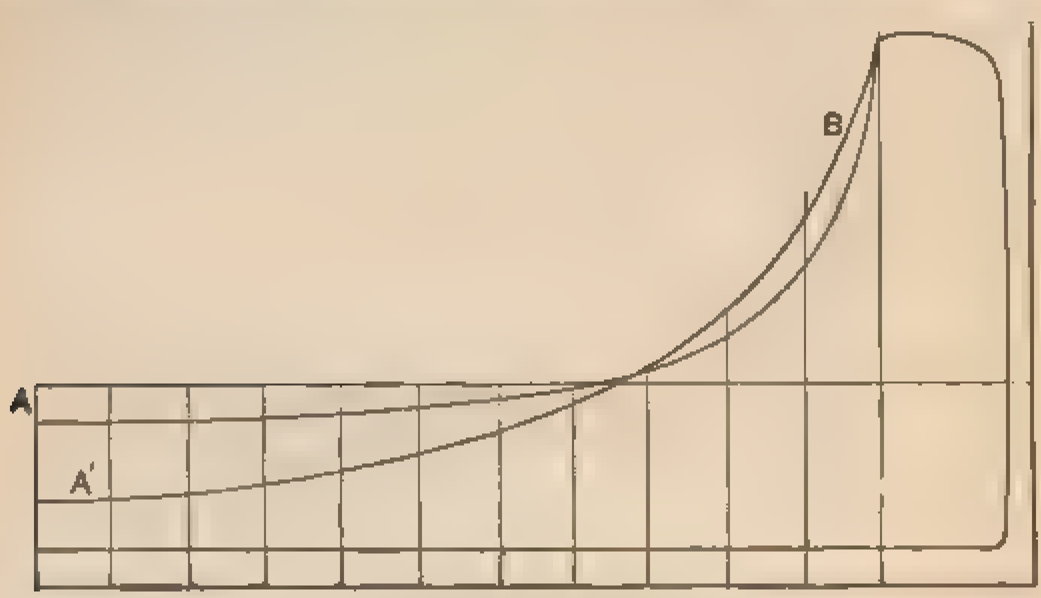

FIG. 79.—CYLINDER CONDENSATION.

taken in throughout the stroke to 2; compression occurs from 2, the pressure rising again to atmospheric at 3 and reaching the

* The Otto Gas-engine, Van Nostrand's Magazine 1883.

limit of the spring at the line 4, 5, 6. The mixture is fired on the succeeding stroke, the pressure continuing above the limit of the spring to 7; then the exhaust-valve opens and the expulsion occurs, producing the line 7, 8, 9, 10, 1. The depressed part 9–10 may be due to inertia.

In the next example a steam-engine moving very slowly gives an expansion BA, which differs remarkably from the curve BA' of Marriotte, usually closely followed by good engines. The cause is now well known to be what is called internal or cylinder condensation, and re-evaporation—a phenomenon discussed fully elsewhere.

CHAPTER VI.

MEASUREMENT OF DIAGRAMS: COMPUTATIONS; APPARATUS AND METHODS.

65. **The Apparatus and Methods** of measurement of the power of the engine by means of indicator-diagrams are necessarily somewhat different with differences of purpose and of data desired. They include such as aid in the direct measurement of the diagrams, and also instruments employed to measure the speed of the engine and its fluctuations. Among the former is the planimeter; among the latter, speed-indicators, counters, and chronographs of various kinds. The methods of their use and of computations based on their work should always be such as will yield results of the greatest practicable exactness.

66. **The Measurements** taken in working of indicator-diagrams demand great care and accuracy. The figure to be measured is small; its bounding-lines often obscure and generally irregular; and the determination of its exact area, which is the usual problem, requires nice manipulations. An indicator-diagram represents the pressures, volumes, and work of the steam, or other fluid, at every instant throughout a single revolution of the engine, on *one* side of the piston. A pair of cards exhibits these quantities on both sides during one revolution. A series of such pairs exhibits the varying pressures and work of the engine at the several single revolutions to which they severally appertain. The average of the pressures shown on one card is the mean pressure for a single revolution on one side the piston; the average obtained from a series of diagrams gives a mean of the pressures, for the period covered, with a degree of approximation dependent upon the number of diagrams and the uniformity of action of the engine. By taking diagrams with sufficient frequency, any desired accuracy may be attained. In practice, they are often taken as seldom

as once an hour, and, on trials of importance, sometimes as often as every fifteen minutes. At sea, it is customary on naval vessels to take a set of diagrams once a day to be preserved in the log-book.

Since the diagram only gives the pressures, the other factors of work and of power must be determined otherwise. The indicated work of the engine at each stroke is the product of the net intensity of pressure on its piston by the volume traversed. The power is the work done by the engine in the unit of time; in British measures,

$$\text{H. P.} = 2\,\frac{plan}{33000};$$

where p is the average net effective pressure of the steam as shown by the indicator; l is the length of stroke; a is the effective area of piston, n is the number of revolutions per minute. Pressures are here, as usual, measured in pounds on the square inch, areas of piston in square inches, the stroke in feet, and work in foot-pounds per minute. Of these quantities, all but p are obtained by direct measurement and by observation. The pressure p is the one quantity obtained by the use of the indicator. The method of determination is to measure the area of the diagram, divide by its total length, and thus obtain in the quotient its mean altitude. This being multiplied by the scale of the spring and of the ordinates gives the mean pressure.

The *mean total pressure* is this quantity measured to the vacuum line of the card. The *mean effective pressure* is the mean pressure measured from the indicator-card, and that which represents the net pressure acting in the production of the indicated work. It is this pressure, usually, which is sought in making computations. The area of the diagram may be obtained in either of several ways.

The best method is by the use of the planimeter, which, with careful handling, should give the area to hundredths of a square inch. Divide the area by the length, and the result will be the height of a rectangle having equal area, and the average height of the actual diagram. Or:

Draw ten, or any other equidistant number, of lines, as in Fig. 80, perpendicular to the atmospheric line. The first and

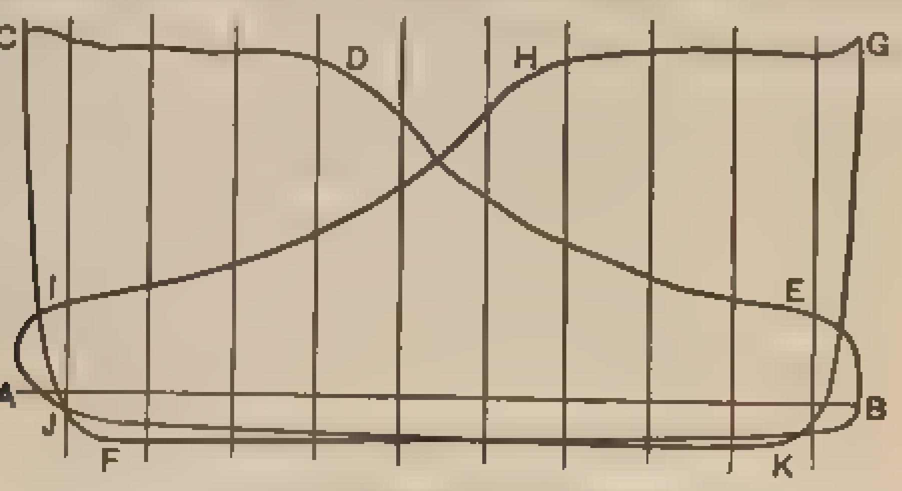

Fig. 80.—Measuring the Diagram.

last half of the intermediate distance from the ends, and the height of each, represent the approximate height of the space which it marks. Measure the length of each ordinate, and divide the sum by the number of ordinates. Multiply the average length thus found by the scale of the spring, and the result is the mean effective pressure.

In case there is a loop, as in Fig. 81, caused by expanding below the back-pressure line, the engine being non-condensing,

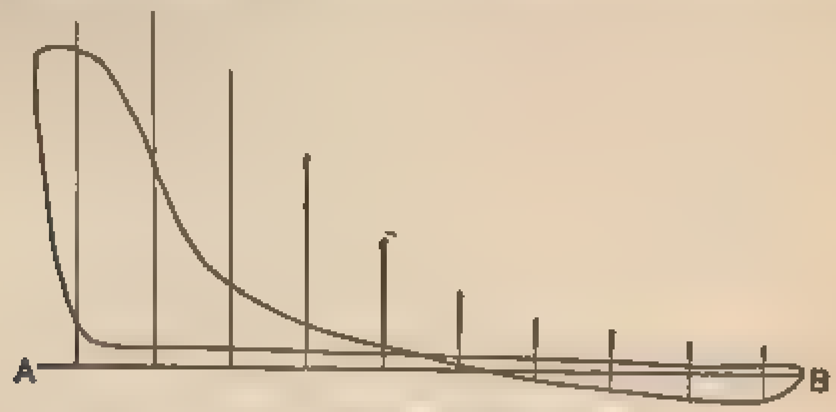

Fig. 81.—Measuring Diagrams.

the ordinates below are negative, and must be subtracted from the lengths of the ordinates above. Then, using the pressure so obtained, multiply the net area of the piston by the mean effective pressure (M. E. P.). Multiply this product by the distance through which the piston travels per minute, divide by 33,000, and then, as already seen,

$$\text{I. H. P.} = \frac{\text{Net area of piston} \times \text{M. E. P.} \times \text{revs. per minute} \times 2 \times \text{stroke.}}{33000}$$

When there are a number of diagrams from the same engine, the calculations may be simplified by multiplying the area of the piston by twice the length of the stroke, and dividing the result by 33,000, thus grouping the constants, and thus obtain the "constant for the engine," the power developed at one revolution per minute with one pound mean effective pressure. If we multiply this constant by the number of revolutions and the mean effective pressure, the product will be the I. H. P. If the number of revolutions is constant, multiply the "*constant for the engine*" by the revolutions per minute. This gives "*horse-power constant*," or power developed per pound of effective pressure. Multiply this constant by the M. E. P., and the result will be the indicated horse-power.

A quick method of measurement of ordinates is to use a strip of paper, and mark, one after another, the lengths of ordinate on its edge, thus making the addition, and with a single final measurement. Lay the edge of the paper on the first line, mark off the distance, starting from the end of the paper; transfer the edge of the paper to the last line, add its length to the first measurement, and continue the addition for the intermediate divisions. Finally, measure, with the scale of the spring, the total length, and divide the result by *ten*.

A small adjustable set of parallel rods is often supplied by the makers with each pair of indicators, which may be used, as in the illustration, to lay off and describe the ten ordinates usually adopted in measuring the diagram. Or a scale dividing off ten parts in a space a trifle longer than the paper may be laid on the diagram diagonally, in such manner that its extremities will coincide with the ends of the diagram and the location of ten ordinates pointed off on the paper, between which the vertical measurements should be made. The use of a scale prepared like that seen in

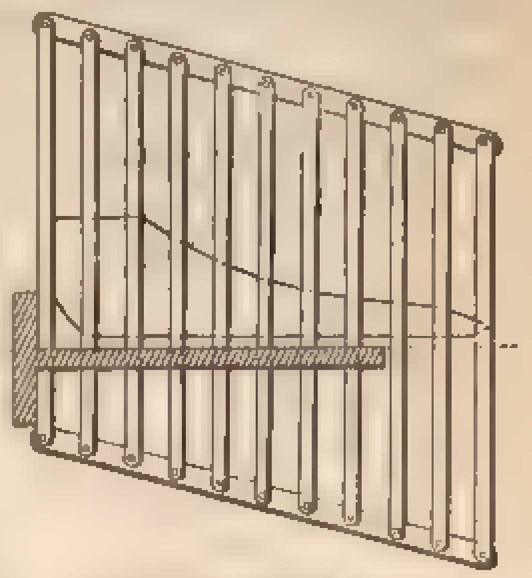

FIG. 82.—PARALLEL RODS.

the accompanying sketch, and which can be easily adjusted to any ordinary length of diagram, is still more convenient. Here the ordinates are correctly set, so as to give a half-space at each end to admit the measurement being made on the lines.

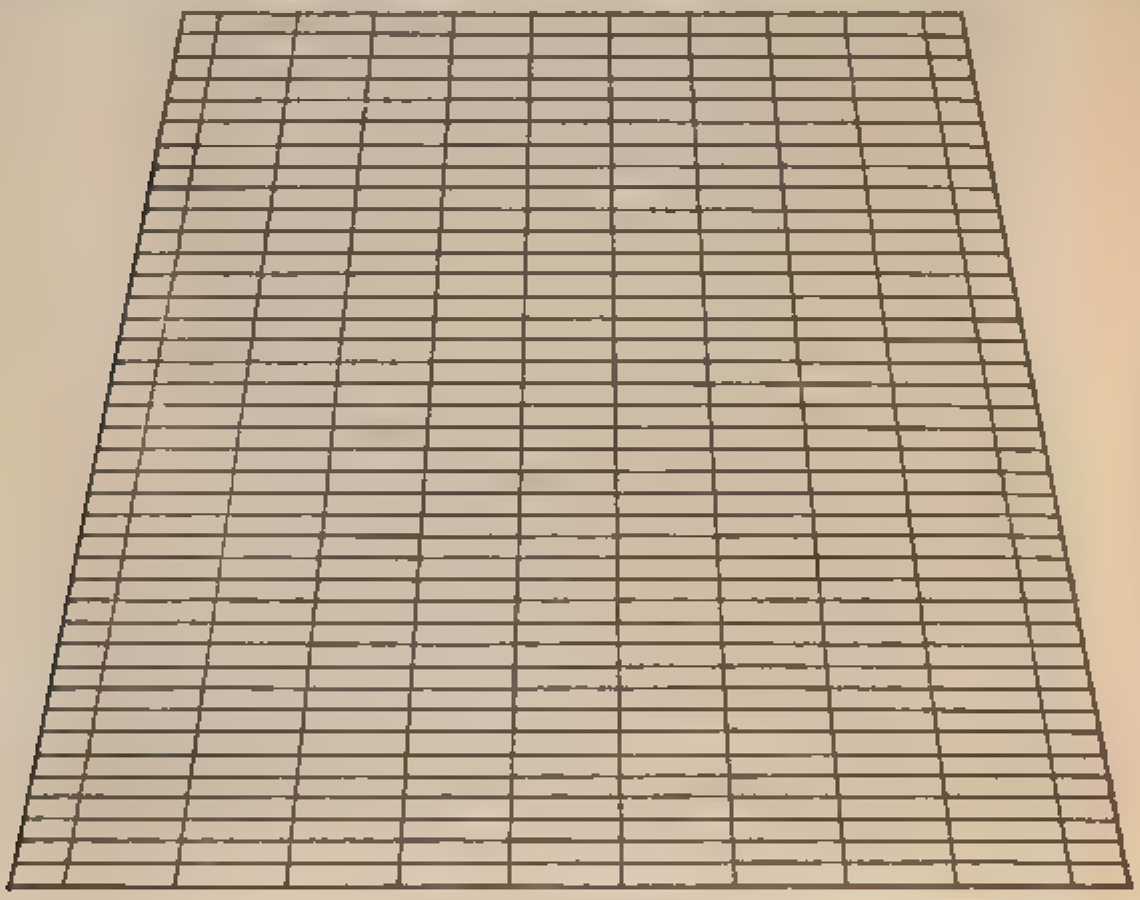

FIG. 83.—SPACING-SCALE.

A very rapid approximation to a correct volume of the mean pressure can be obtained by the expedient illustrated in Fig. 84.

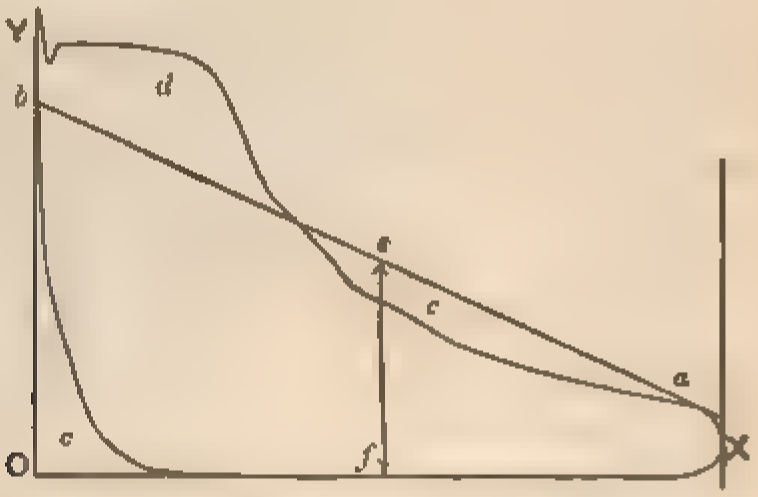

FIG. 84.—MEAN PRESSURE.

Ox being the back-pressure line, draw *ab* in such manner as to make the areas *c* and *d* between that line and the upper border of the card as nearly equal as possible. This can be quite closely judged by the eye. Then the ordinate, *ef*, drawn at the middle of the diagram gives the mean effective pressure.

The following, from Rankine, illustrates a good method of record and computation.* Ten ordinates are measured and the results for both cylinders of a compound engine are given.

COMPOUND-ENGINE DIAGRAMS.

Ordinates.	First Cylinder.		Second Cylinder.	
	Top.	Bottom.	Top.	Bottom.
b_0	27	36	16.0	12.4
b_{10}	13	12	2.0	3.8
Sum	40	48	18.0	16.2
Half sum	20	24	9.0	8.1
b_1	83	97	10.5	10.8
b_2	91	96	8.5	9.0
b_3	91	84	7.5	8.0
b_4	64	64	7.0	7.1
b_5	57	57	6.6	6.7
b_6	53	46	6.2	6.0
b_7	42	40	6.0	5.6
b_8	35	32	5.1	5.4
b_9	22	22	4.5	5.0
Sum	558	562	70.9	71.7
Sum ÷ 10 = M. E. P.	55.8	56.2	7.09	7.17
Mean, top and bottom	56.0		7.13	
× area of piston, sq. ins.	345		1380	
Mean effort, in lbs	19320		9839.4	
× stroke, in feet, 2½ × revolutions per minute, 52½ × 2 =	262.5		262.5	
Ft.-lbs. per minute	5071500		2582842.5	
Total	7654342.5			
÷ 33000 =	232 I. H. P.			

* Steam-engine, p. 51.

These mean pressures are found by a process which may be algebraically represented as follows:

Divide the length of the diagram into any convenient number, n, of equal parts, and measure the ordinates at the two ends and the $n-1$ points of division; so that ordinates are measured from $n+1$ equidistant points.

Let p_0 be the first, p_n the last, and p_1, p_2, etc., the intermediate ordinates of the upper curve; let p_0' be the first, p_n' the last, and p_1', p_2', etc., the intermediate ordinates of the lower curve; let p_m denote the mean forward pressure, p_m' the mean back pressure, and $p_m - p_m'$ the mean effective pressure. Then

$$p_m = \frac{1}{n}\left(\frac{p_0 + p_n}{2} + p_1 + p_2 + \text{etc.}\right)$$

$$p_m' = \frac{1}{n}\left(\frac{p'_0 + p'_n}{2} + p'_1 + p'_2 + \text{etc.}\right);$$

$$p_m - p_m' = \frac{1}{n}\left(\frac{p_0 + p_n}{2} + p_1 + p_2 + \text{etc.} \ldots - \frac{p'_0 + p'_n}{2} - p_1 - p_2 - \text{etc.}\right).$$

The mean effective pressure may be computed at once by measuring a series of equidistant breadths of the diagram the mean of which breadths represents the mean effective pressure. That is, let b_0 be the first, b_n the last, and b_1, b_2, etc., the intermediate breadths.

Then

$$p_m - p_m' = \frac{1}{n}\left(\frac{b_0 + b_n}{2} + b_1 + b_2 +, \text{etc.}\right).$$

The effective energy exerted by the steam on the piston during each revolution is twice the product of the mean effective pressure, the area of the piston, and the length of stroke, or

$$2(p_m - p_m')As;$$

and if N be the number of double strokes in a minute, the indicated power in foot-pounds per minute is

$$2(p_m - p_m')ANs;$$

from which the indicated horse-power is found by dividing by 33,000.

The presence of the piston-rod or of a "half-trunk" on one side the piston produces a difference of areas which, in the latter case, is of considerable magnitude. Where measured separately, if A_1 and A_2 are the two areas, the power is

$$\text{I. H. P.} = \frac{(p_1 A_1 + p_2 A_2)ln}{33000},$$

l and n being the length of strokes of piston and the number per minute.

67. Planimeters, plane-area measuring instruments, mechanical integrators, as they are variously called, furnish the best known means of measuring the area of the indicator-diagram. By their use the work can be done by an expert hand with great rapidity and with marvellous accuracy. Liability to error is exceedingly small, and the magnitude of the probable error is quite inappreciable in the ordinary work of the engineer *—sometimes as little as one part in above ten thousand. Errors exceeding one tenth of one per cent are usually due to inexperience of the operator. The best known instrument is that of Amsler; that of Coradi,† of similarly general application, and that of Coffin,‡ designed especially for the measurement of indicator-diagrams, are also in common use, the two former mainly in Europe, the latter in the United States. They commonly operate by the combined sliding and rolling motion of a small measuring wheel which has a total rotation proportional to the area enclosed by the figure the periphery of which it traverses.§ The integrating wheel, or roller, is best made of steel. A vernier on the instrument en-

* "Ueber die Genauigkeit der Planimeter," Professor Lorber, Oesterreiche Zeitschrift für Berg- und Hüttenwesen, vol. xxxi; p. 22.

† Ueber das Roll Planimeter von Coradi; Franz Lorber, Zeitschrift des Oesterr Ing. & Arch. Verein; vol xxxvi; p 135.

‡ Barrus on the Indicator, p 61.

§ Bramwell on the Amsler Planimeter. Report Brit. Association, 1872, p. 401; Shaw on Mechanical Integrators; Proc. Brit. Inst. C. E., 1884–5, No. 2063.

ables the readings of the motion of the roller to be taken with great accuracy, and repeated measurements may be made to eliminate errors less in amount than the finest reading given. A very simple modification of the Amsler Planimeter, designed by Mr. J. W. See, is made especially for indicator work.

The following are the maker's instructions, in detail, for using the most recent form of Amsler planimeter:

1. Adjust the screw-centres upon which the index-roller D revolves, so that the roller works freely, and does not touch the vernier. The same care must also be taken with the centre-pin C. Oil the screw-centres now and then. Care should be

FIG. 85. AMSLER PLANIMETER.

taken to prevent the tube B, the tracer F, and the point E from being bent, and also to see that the barrel D is kept uninjured.

2. To find the area of any figure, set the roller D and the counting-wheel G to zero; the square rod A must be pushed into the tube B, and the line on A marked 1 sq. dem., or 0.1 sq. ft., etc., must come even with the small line on the bevelled part of the tube B; when this is done, place the instrument on the paper, and see that the roller D, the tracing-point F, and the needle-point E touch the paper. Press the point E slightly into the paper, and put the small weight on the hole over the point; the instrument is then ready for work.

3. Take any point P on the outline of the figure about to be measured, set the tracing-point F to that point, and when it is marked, read off the index-roller D and counting-wheel G. For example, suppose the counting-wheel shows 2, the roller 91, and the vernier 5, the number will be 291.5. Follow the

outline of the figure with the point F as accurately as possible to the right, until you come to the starting-point. Straight lines can be followed along a ruler; then read off the numbers on wheel and roller; say it is the second time 476.7.

4. When these two numbers are obtained, there are two cases to be observed:

1st. If the point E is outside the figure, subtract the first reading 291.5 from the second 476.7, the remainder is 185.2, which shows that the area contains 185.2 units. Of course the units depend entirely on the regulation of the bar A, if they are 0.1 sq. ft. we have $185.2 \times 0.1 = 18.52$ sq. feet, as the area of the figure measured on the paper.

The rule therefore is, when the point E is outside, multiply the difference of the two readings by the number on the bar to the right of the corresponding division.

2d. When the point E is inside the figure, before making the subtraction, the number engraved on the top of bar A, above the corresponding line of division, must be added to the second reading. In this instance, suppose the number on top of bar A is 20.985, the second reading is 4.767, the calculation would be as under:

Second reading	=	4.767
Number over 0.1 sq. ft.	=	20.985
		25.752
Deduct first reading	=	2.915
Remainder		22.837

The two points seen on the bar being set to the exact length of diagram, the readings are in I. H. P. for an indicator-scale of 40 pounds to the inch.

In using the Coffin planimeter, the diagram is set with the atmospheric line parallel with the lower edge of the clamp C, and the end even with the perpendicular edge. The clamp K is moved up to the other end. The block Q being introduced into the groove I, the tracer is set upon the point D, where the edge of the clamp L touches the figure. The wheel is

turned so as to bring the reading to zero, and the tracer is then moved over the line of the diagram in the direction of the motion of the hands of a watch. The tracer is then moved along the edge of the clamp till the reading is again made to

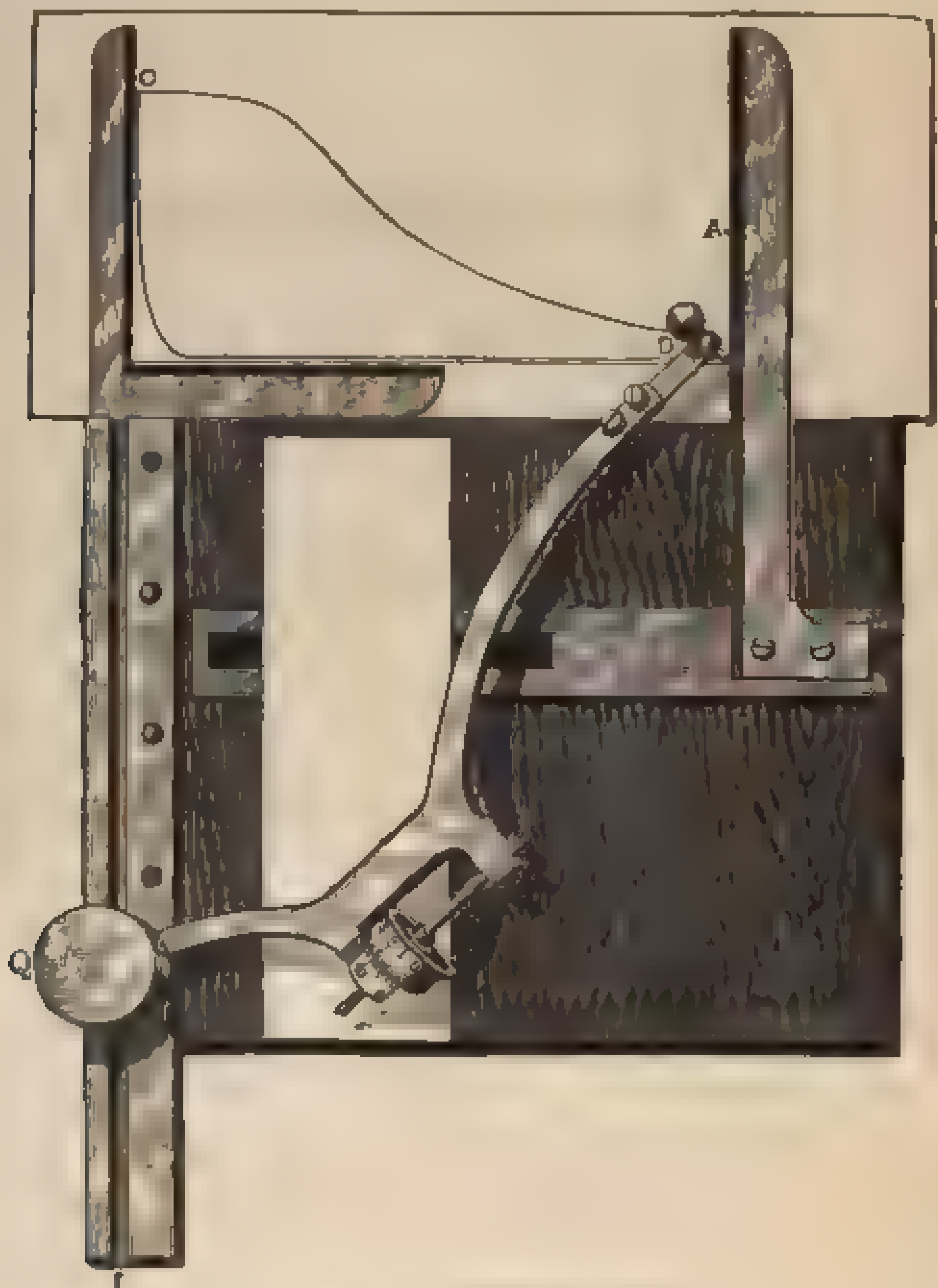

FIG. 86.—COFFIN PLANIMETER.

zero. The distance of *A* from *D*, now measured on the scale of the spring, is the mean effective pressure. The reading on the wheel is the area of the diagram in square inches. From this area the mean effective pressure may be also computed,

by multiplying it by the quotient obtained by dividing the number of the spring by the length of the diagram in inches.*

This measurement may be made and computations computed on forty or fifty diagrams per hour by an expert computer using this instrument, obtaining the value of the mean effective pressure, inserting it in the formula already given, and computing the indicated horse-power.

Mr. Lea has made an interesting modificator of the ordinary indicator by the substitution of a planimeter for the pencil-motion,—either permanently or temporarily, as may be desired, thus at once getting a measure and registry of the work done.†

68. Tachometers, Speed-indicators, Chronographs, and Counters are instruments of various kinds and classes used by the engineer in determining the speed of the engine, the second of the essential factors obtained by observation in measuring its power. Of these instruments, the counters and many speed-indicators give the exact and positive measures of the engine-speed required by the observer as data. They mechanically register the revolutions of the machine one by one, and give either the totals or the differences for selected intervals; or, more usually, they work continuously, and these totals or differences are read off at regular intervals by the observer and recorded in the log, thus giving the means of obtaining the average speed of engine throughout the trial. When indicator-diagrams are taken, the speed of rotation of the engine is taken as nearly simultaneously as possible. This measurement is commonly taken with one of the small hand speed-indicators, the spindle of which is applied to the centre in the end of the main shaft and there held a quarter or half minute, a full minute, or more, as less or greater accuracy is desired and as the speed of the engine is greater or less. Very exact measurement is usually demanded for purposes of computation.

The "tachometers" are a class of instruments which exhibit

* Barrus, p. 63.

† Mechanics; Jan. 20, 1883; p. 39.

on a dial the speed of rotation of the shafts by which they are driven. They are not generally relied upon to give exact readings; but their closely approximate indications check the hand-counter record at any moment, as the hourly or daily readings of the mechanically registering counters permanently attached to the engine check the averages of that record. The tachometers, Fig. 87, are actuated by pulleys or gearing, and are designed to indicate the number of revolutions performed per minute by shafting, by a pointer travelling over a graduated dial. In the cylindrical case rotate two suspended weights or pendulums, connected together by a strong flat coiled spring. The purpose of this spring is to counterbalance the centrifugal force of the pendulums. The deviations of the pendulums are communicated by a rod to clockwork in the case behind the dial, and produce corresponding deflections of the pointer. These instruments possess the advantages of exhibiting to the eye the momentary fluctuations of speed which cannot be thus shown by the revolution-counters. In some cases, by the addition of a recording mechanism and a roll of paper to receive the record, the tachometer is converted into a "tachograph," and it is in this form often attached to engines or other machines to supply a constant and permanent record of their operations. For experimental purposes the paper is driven at comparatively high speed, as high as an inch (2.5 cm.) per minute, the more common speed being one half or one quarter that velocity. When used on the locomotive engine, it is customary to mark the dial in miles per hour as well as revolutions per minute. Various forms of the instrument are made for the various purposes of engineering, and are applicable to all speeds—from that of the slowest engine to that of the fastest electric machinery. The Edson speed-recording instrument, and the Mosscrop and other familiar apparatus, are of this class.

FIG. 87. TACHOMETER.

The modern Speed-recorder is an instrument, Fig. 88, which registers the fluctuations of speed of the engine, or of other machinery, on a travelling strip of paper actuated by a clock. The variations of velocity are produced by the movement of a revolving pendulum, like the Watt governor, which moves a pencil across the line of motion. The curve thus traced is a record of the whole history of the time of observation. A widely serrated line shows great irregularity of speed; the less and the closer the serrations, the better the speed; the rise of the line above the mean indicates a steady acceleration; a fall means retardation; a wide, even, band of oscillations shows a light wheel; a narrow band along the correct mean indicates a good balance-wheel; a gradual fall may indicate change of steam-pressure, and sudden variation of location of the record line or band usually indicates fluctuation of the load, and its extent of fluctuation the effectiveness of the system of regulation. The kinds of unsteadiness due to changing load and pressure and to inefficiency of regulation are easily distinguished, and the various causes of variation of speed may usually be easily traced out and remedied.

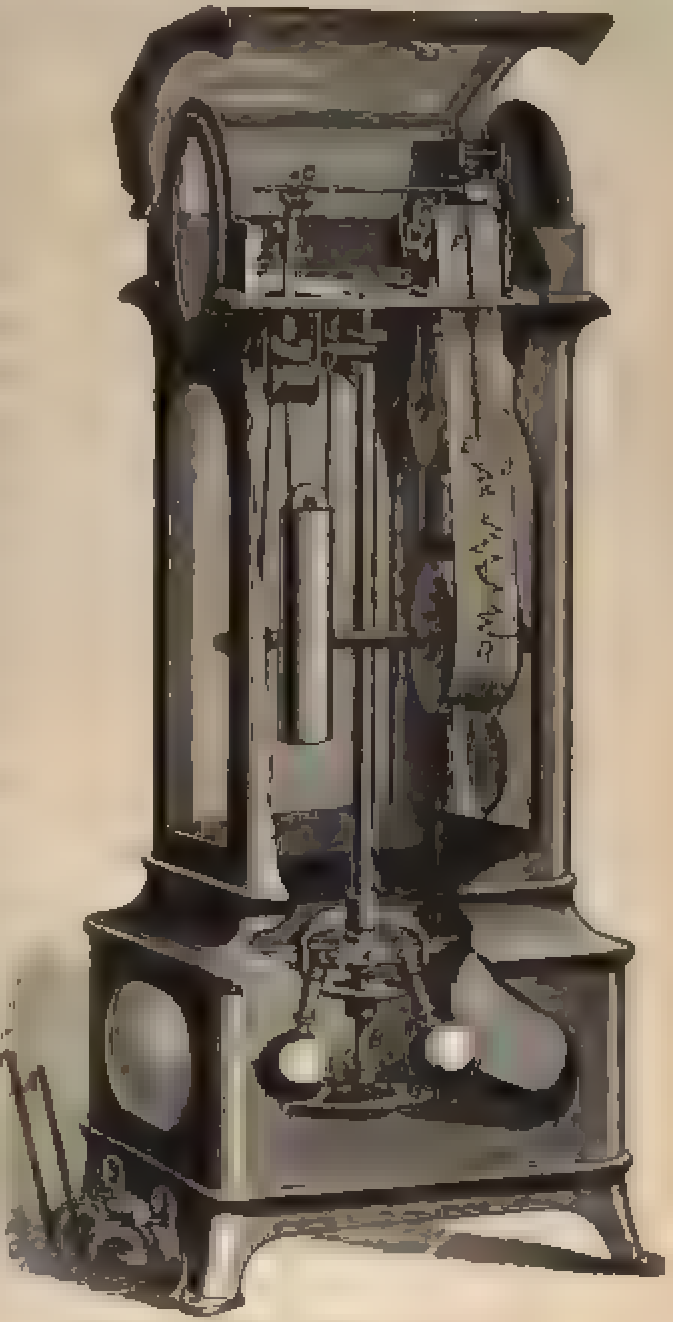

FIG. 88.—THE MOSSCROP RECORDER.

Chronographs, such as are employed by the physicist, may be used when it is desired to determine the method and extent of variation of speed during any single revolution of the engine,

or part of the stroke, as in investigating the effect of the varying pressure on the piston and the torsional moment at the shaft · conditions which can only be studied experimentally by the use of apparatus of extraordinary delicacy and quickness of action and record.

The chronograph was first applied to measure the variations of velocity of the steam-engine by a committee of the British Association in 1843-4, in their determinations of the speeds of piston of the Cornish engine.* It has been applied by Mr. Woodbury, as early as 1873, to pumping-engines, determining the fluctuations of velocity of fly-wheel, and, later, by Mr. Eckart in observing similar fluctuations of pump-rod speed, at great depths in the Comstock lode in Nevada.† The latter found it practicable to use the instrument at speeds of from 80 feet per minute up to 1400 feet. Messrs. Dix and Mack, under the supervision of the Author, applied the same instrument, still later, to the "high-speed" engine, making 250 revolutions per minute.

The following are Mr. Woodbury's records from three engines: ‡

Portion of Revolution	Lowell Pumping-engine	Lynn Pumping-engine		Horizontal Engine
	Revolutions per Minute			
	13.26	18 61	13 90	19.39
	Velocity in Feet per Second			
.00	6 42	9 80	7 13	10.16
.04	6.46	9 82	7 27	10 62
.08	6.54	9 92	7 53	10 84
12	6.68	10 08	7.72	10 96
.16	6 84	10 14	7 75	10.90
.20	6.96	10.16	7.70	10 72
.24	7 06	10.10	7 53	10 42
.28	7.10	9.96	7.33	10 04
.32	7 06	9.70	7 02	9 58
.36	7 02	9 43	6 60	9 25
.40	6 92	9 23	6 20	9 14
.44	6 82	9 08	5 87	9 27
.48	6.77	9.00	5 73	9.06
.50	6.75	8 97	5 71	9 92

* Trans B A A S 1844

† Trans Am. Soc M E, Vol. III, 1882

‡ Ibid.

The curves here shown represent the motion of the fly-wheel of the Lynn engine.

By the use of the ordinary form of physicist's chronograph, or a slightly modified instrument, the speed of engine, and its variations, are measured, not only stroke by stroke, but even from point to point in the single revolution of the engine. This is a matter of importance in the application of engines, and especially if at low speed, to the driving of dynamo-electric

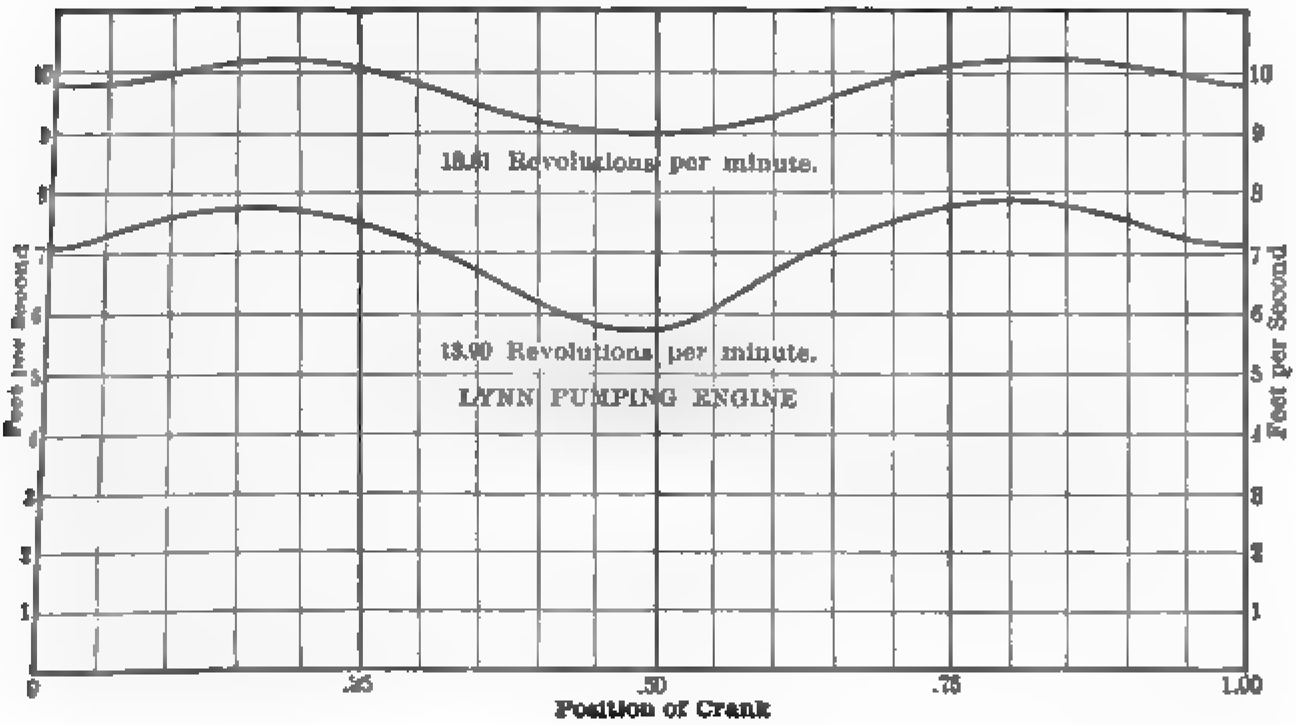

FIG. 89.—MOTION OF FLY-WHEEL.

machinery, where variations of speed, however limited as to time, are seriously objectionable. The frontispiece exhibits the method of attachment of the instrument to a "high-speed," direct-acting, horizontal engine of common type. *A* is the steam-cylinder; *B*, the engine-frame; *C*, the end of the main shaft; *D*, the balance-wheel; *E*, the brake-pulley, with strap *F*, and scale weighing the turning effort at *G*. On the extremity of the shaft, a coupling, *H*, is attached which drives the chronograph, *I*, through a slender rod seen connecting them. The revolving cylinder, on which the paper is smoothly stretched, to receive the record, is seen at *J*, and the stylus or pen is at *K*. The whole is mounted on a firm support as *L*.

When in operation, the cylinder is turned by the engine,

instead of its usual motor apparatus, and the pen, slowly traversing the cylinder, produces a closely compressed helix. At regular intervals, a circuit is made and broken by the standard clock or other timing instrument, and the line is given a V-shaped jog which marks the time-interval on the cylinder. The adjustment should be such that, at normal speed, these breaks should occur at precisely the same points in the circumference of the chronograph cylinder at each of its revolutions or at each tenth or other fraction of a revolution, as may be determined upon. Any acceleration or retardation will then be exhibited by the production of the break in advance of, or behind, its normal position. In the first case all such breaks fall into straight lines along the cylinder, parallel to the axis; in the latter case they will fall into regular helical, or curvilinear, or irregular, lines, accordingly as the acceleration or retardation is uniform, smoothly variable, or irregular. The inclination of the lines, or of the tangents to the curves so produced, to the axis is thus a measure of the change of speed.

Thus, if

C = the circumference of the record cylinder;
d = distance traversed by the pen per revolution;
n = number of revolutions of engine per minute;
n' = revolutions of the chronograph;
$c = \frac{n}{n'}$;
θ = angle made by the line produced as above with the axis,—

then we have

$$\frac{Cn}{60c} = \frac{C \pm d \tan\theta}{2n'}, \quad \text{and} \quad n = \frac{30c}{n'}\left(1 \pm \frac{d}{C}\tan\theta\right);$$

using the positive and negative signs for acceleration and for retardation, respectively.

When $\theta = 0$, $\qquad n = \frac{30c}{n'}$.

For a case of actual work, $C = 21.84$; $d = 0.0833+$; $r = 285$; $n = 30$; and

$$n = 285\left(1 \pm \frac{d}{C} \tan \theta\right);$$

and, making a variation of $1°$, the angular deviation would become $\tan^{-1} \theta' = 42° 36'$; since

$$n - 285 = \frac{nd}{C} \tan \theta' = 1,$$

and

$$\tan \theta' = \frac{C}{nd} = \frac{0.833+}{21.84}; \tan^{-1} \theta' = 42° 36'.$$

It often requires very nice adjustment to secure sufficiently perfect arrangement of speeds to give a good line for the normal operation of the engine; especially as the sensitiveness of the instrument increases with decreasing values of the angle θ.

The following is a set of data thus obtained in the course of a trial of a good type of engine, considerably out of adjustment :

Observation.	θ	$n_1 - n_2$	n_2	*D. H. P.*
1	$+ 75° 51' 80''$	+ 4.35	289.35	0
2	$- 72° 14' 17''$	− 3.55	281.45	4.22
3	$- 78° 32' 58''$	− 4.54	280.46	7.01
4	$- 79° 48' 46''$	− 6.05	278.95	9.77
5	$- 80° 43' 54''$	− 6.63	278.37	12.53
6	$- 85° 18' 45''$	− 13.26	271.74	14.95
7	$- 85° 58' 47''$	− 15.47	268.53	17.45

The next engraving exhibits the chronograph as used by Mr. Eckart. The reference-letters are as follow:

CC—Cast-iron base-plate, covered with sheet brass, upon which the mechanism is secured.

B—Metal frame containing gearing for driving drum *A* and escapement-wheel *b*; motion communicated by means of adjustable weights *D*.

AA—Light brass drum, accurately balanced, revolving on friction-rollers 8, 8, at both ends.

f—Parallel guide-bars upon which the tracing-point h_0 and its carriage travel back and forth, receiving motion in one direction from the engine or other moving parts through the cord P, passing through the bars f, and attached to the tracing carriage; the return motion is derived from a coiled spring in the spring drum C.

e—Small electro-magnets on tracing carriage for raising the tracing-point h_0 off the paper and replacing it at any desired point to be especially observed.

d—Electro-magnets on separate carriage kk, adjustable on parallel bars f, operating the steel tracing-point g, at-

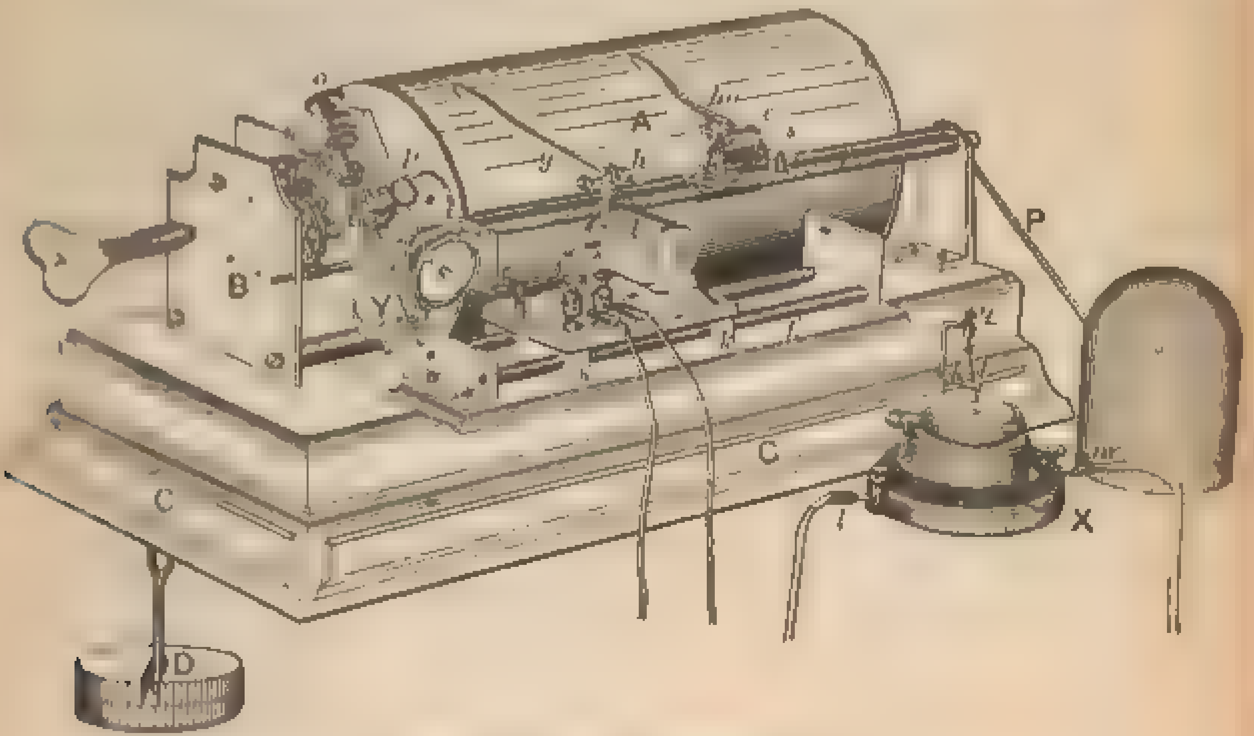

FIG. 90.—THE CHRONOGRAPH

tached to the armature of d, for the purpose of recording seconds on the margin of the paper or at other parts of same as required.

t—Chronoscope or watch supported on frame x, the second-hand of which swings the light platinum wire J, breaking contact with the insulated wire k, thereby breaking circuit with d and recording seconds through the tracing-point g on the paper.

q—Adjusting screw for the wire J.

a—Steel spring of escapement. This spring is securely

clamped in *Y*, its flexibility being controlled to a certain extent by means of the thumb-screws *o* and *p*.

This instrument was found to give with great exactness the fluctuations of piston-speed in a pumping engine at 80 feet per minute, and for a hoisting engine at 1400 feet. It is thus peculiarly well adapted for pit-work.

Instead of, as is usual, employing a clock pendulum to mark time and give the velocity-curve of the engine, a portable time-keeper is here used. This is the common form of "timer," designed especially for timing in trials of speed of vessels, or on the race-course. The hand of the instrument, revolving once per second, breaks the circuit, and the stylus or pen *g* is caused to mark the interval. The stylus or tracing-point barely touches the lamp-black, being counterbalanced in such manner as to remove the coating without bearing perceptibly upon the paper, producing a fine white line on the black surface. In fitting the paper, it is cut slightly longer than the circumference which it is to cover, and lapping the edges and gluing them together, the lap is carefully sandpapered to as nearly as possible uniform thickness at the joint and elsewhere. The surface is then smoked, and is ready for use.

The Hand Speed-indicator is made in various forms. One which the Author has found very satisfactory is shown in the illustration. It answers equally well in whichever direction the engine may turn, is convenient in use, and gives reliable results. The Author has often found a different shape of point more certain to hold, and has often flattened the point and given it a semicircular sharp-edged termination, to obtain a more secure hold on the centre of the shaft. (Fig. 91.)

The usual method of counting the revolutions of the engine, by means of the hand speed-indicator or the registering "counter" attached to the machine, gives the mean speed for a certain time—as for a minute—for which the count is taken. The use of the chronograph in the manner just indicated gives a measure of the *rate* or speed for the instant, for each revolution. To ascertain the rate during successive *portions* of the revolution, the method of Woodbury or of Eckart may be adopted.

These processes may all be used—properly fitting up the chronograph for the purpose, but a much less costly, more

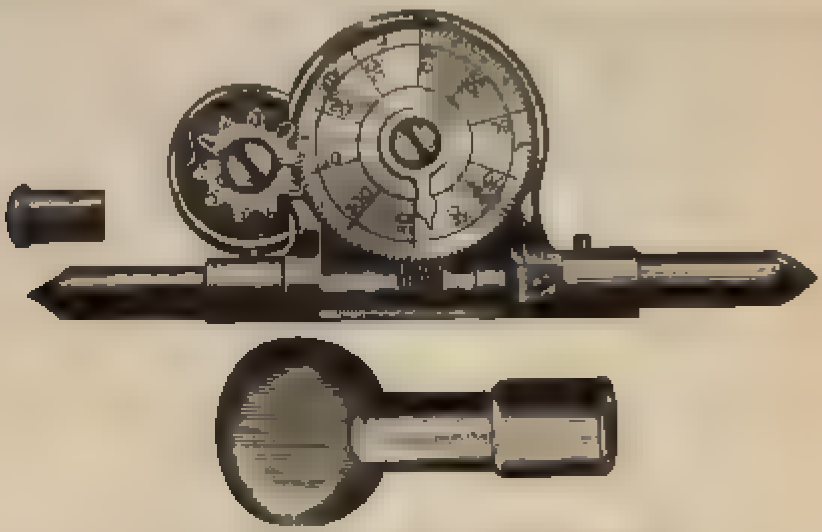

FIG. 91.—DOUBLE POCKET SPEED-INDICATOR

convenient, and simpler method may be employed, the measuring instrument being the ordinary tuning-fork or "timing-fork."

The Timing-fork used in timing engines, and in measuring speeds and speed-variations during a single revolution of the engine, may be of any convenient size, but preferably one of slow vibration and low note. The fastest engines ordinarily make about one revolution in the fifth of a second; but very small engines, and especially those used in electrical and high-speed torpedo-boat work, sometimes make a revolution in the tenth of a second. A standard C fork making 256 vibrations per second would thus space off the engine-cycle into from 25 to 50 parts with the fastest of small engines, or into 256 parts if the engine makes 60 revolutions per minute. The normal fork, for concert pitch, at the Conservatoire de Musique, Paris, makes 870 vibrations for standard *a*, treble staff, corresponding to 261 vibrations for C. This would similarly measure speeds at intervals of 87 or 174 parts for the fastest engines, or 870 parts at 60 revolutions.

The usual method of application is that of Mons. Duhamel, who covers an accurately turned cylinder with a sheet of paper having a smooth and firm surface, uniformly covered with lamp-black, and permits a fork mounted on a firm stationary support to record its motions in a sinuous curve on the paper, as the cylinder is regularly turned beneath the point of a light stylus fixed on the end of the fork. The rate of the fork being

known, and the number of sinuosities of the line being counted for any specified period, the time becomes known. If the line be marked at each revolution or specified part of a revolution of the engine, by any convenient automatic system, the velocities become known for each of those periods. The cylinder carrying the paper should be caused to traverse longitudinally by the action of a screw of conveniently chosen pitch, as is illustrated by the recording mechanism of the Scott phonautograph.* The record on the smoked paper being made, it may be saved and rendered permanent by dipping it in an alcoholic solution of shellac, or of sandarack or other gum.

In engine-testing the following method has been found to be very satisfactory: A toothed wheel or disk is mounted on the end of the main shaft, the number of teeth being determined by the degree of accuracy sought, as 36 to give measures for each ten degrees in the revolution of the crank and shaft, or 72 for five-degree intervals. These teeth constituted the circuit-breaking apparatus for a small battery, the current from which was the primary for a small induction-coil, the induced current being caused to pass through the stylus and to the paper-cylinder, each spark breaking through the smoked surface and leaving its mark, and the space and time between successive points thus made giving a measure of the speed of the engine in that ten-degree interval. Some care is necessary to get a good form of stylus for this work. The sketch shows that adopted by Messrs. Dix and Mack for their investigation of this subject. A light metal frame, *AA*, carries the very fine and light needle, or stylus, *B*, of which the point *P* is smoothly rounded so as not to tear the paper, and which is guided by an opening, *C*, and held up to a gentle contact by the spring of *D*. A small screw, *S*, holds the whole firmly in place on the end *F* of the fork.

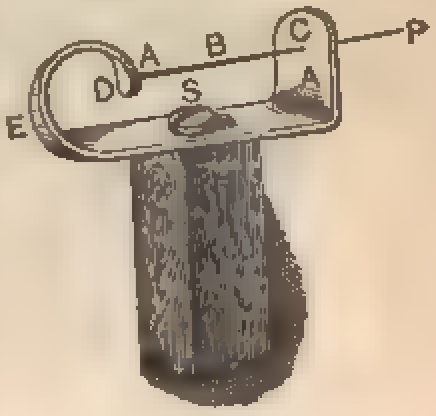

FIG. 92.
STYLUS FOR TIMING-FORK

* See Ganot's Physics, § 246.

The next figure shows an improvised and simple, but effective, method of moving the fork. *B* is the engine-frame; *C* is

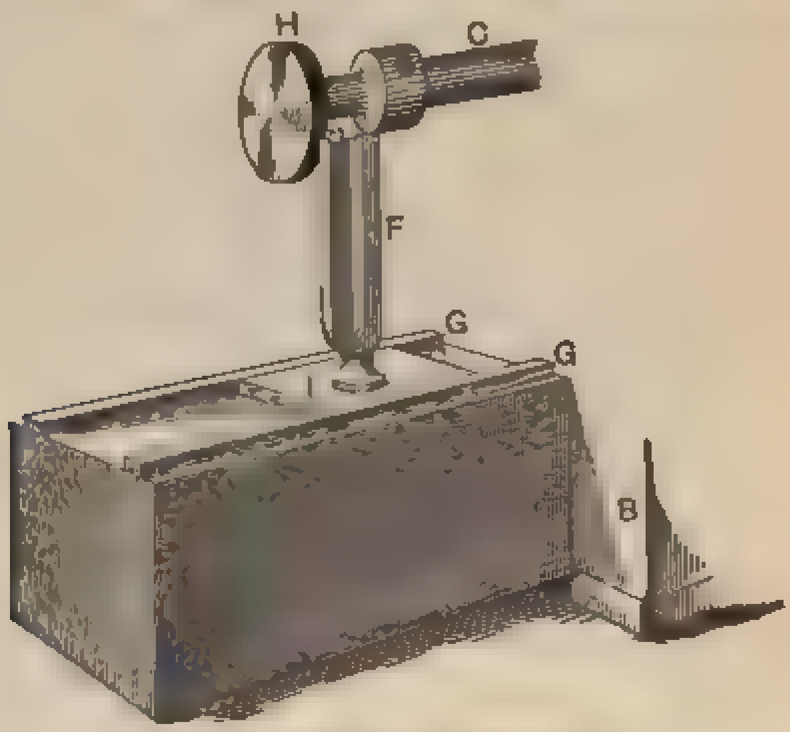

FIG. 93.—MOUNTING THE TIMING-FORK.

the crank-shaft; *F* the fork, mounted on a platform, *GG*, in such manner that it may be smoothly and steadily traversed

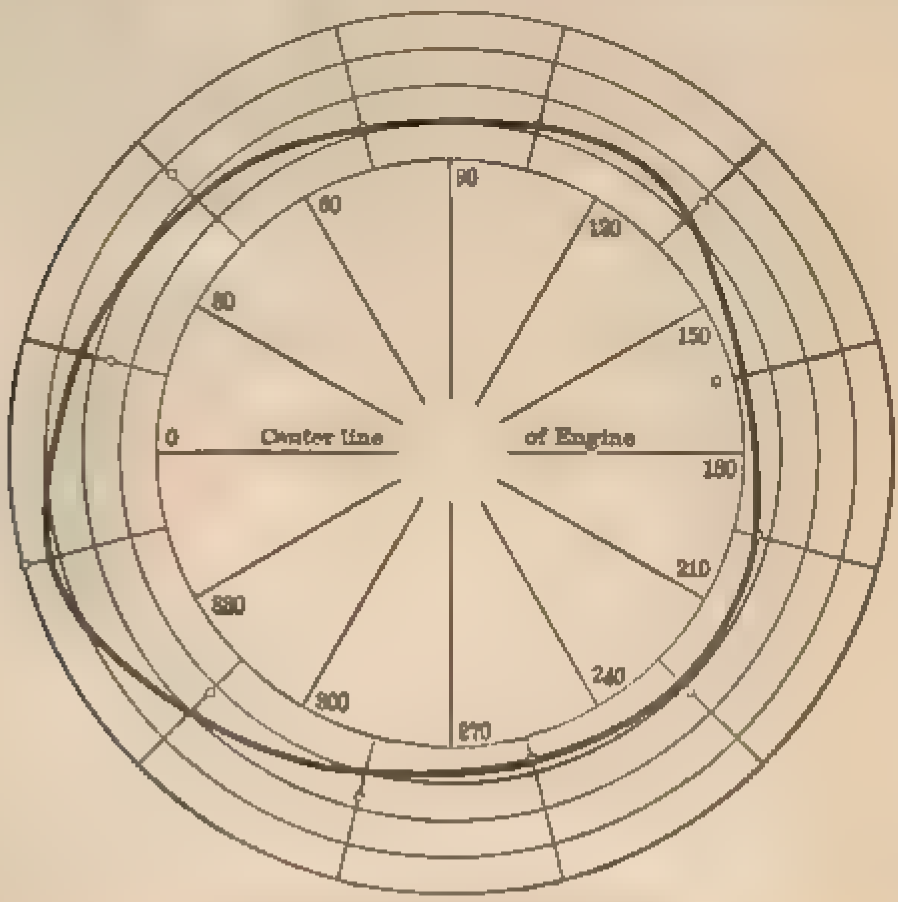

FIG. 94.—VARIATION OF ROTATION.

before the smoked cylindrical surface *H* by sliding its base-piece, *I*, between the guides.

Below are the data thus secured by test of an engine making about 285 revolutions per minute :

Angle	Vibrations.	Variation Per Cent	Angle.	Vibrations.	Variation, Per Cent
0 -30	8.7	−2.3	180 -210°	8.8	−3.4
30 -60	8.6	−1.2	210 -240	8.75	−2.9
60 -90	8.75	−2.9	240 -270	8.8	−3.4
90 -120	8.75	−2.9	270 -300	8.7	−2.3
120 -150	8.7	−2.3	300 -330	8.75	−2.9
150 -180	8.9	−4.6	330 -360	8.5	0 normal.

Plotting these data, the accompanying figure is obtained, and a comparison of the curve with that representing the varying accelerating moments acting on the engine-shaft; the two are found to accord—as should be expected. Radii here measure velocities. The connecting-rod was here six cranks in length, the ratio of expansion about four.

The accompanying illustration exhibits the mounting of the

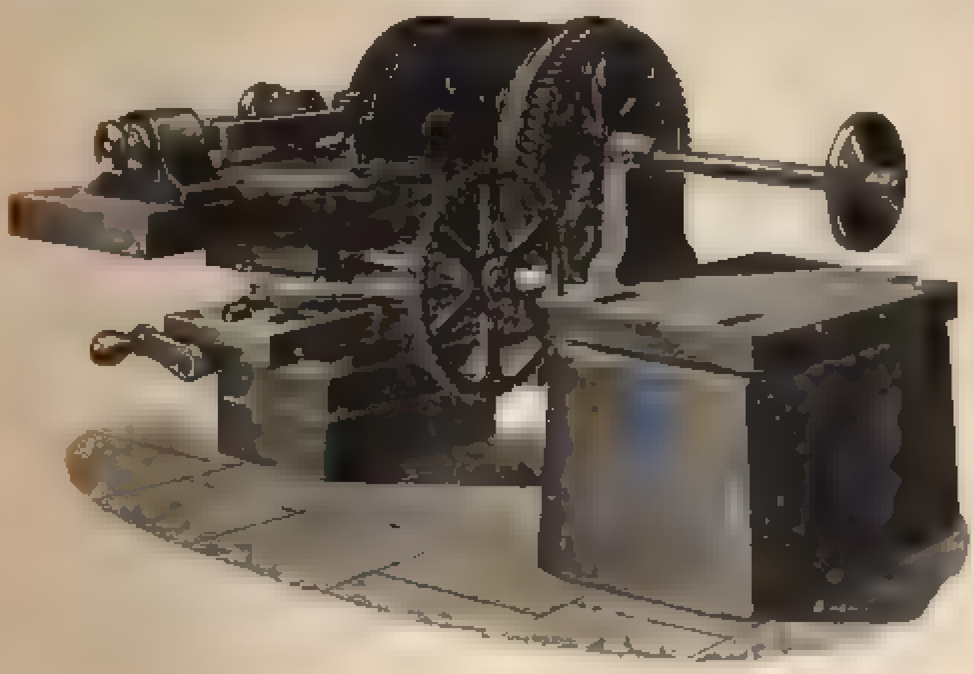

FIG. 95.—TIMING-FORK.

timing-fork as devised by Mr. Ransom.* The timing-fork, kept in motion by an electric current, is mounted on a rest driven by a screw, parallel to the axis of the paper-cylinder.

* Journal of the Society of Arts, Feb. 15, 1889, p. 243.

The operation of the instrument is the same as in the cases already described. The record obtained is similar, and may be illustrated by that given, full size, in the next figure, which represents a mean speed of 141 revolutions per minute.

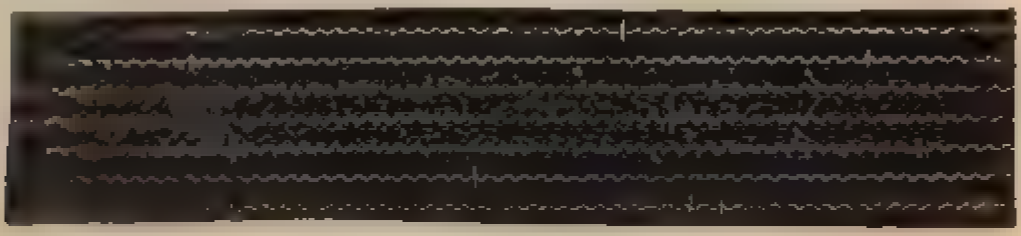

FIG. 96 SPEED RECORD.

Good regulation is usually considered to imply:

(1) Uniform rotation; meaning minimun variation of angular motion during the revolution of the crank-shaft.

This variation depends, for its amount, upon the simultaneous variations of effort and resistance, and upon the magnitude of the regulating mass, the fly-wheel.

(2) The speed of engine, revolution by revolution, should be very nearly constant.

This variation should not usually be allowed to exceed two, and is often less than one, per cent.

(3) The mean engine-speed should remain constant over the whole period of its operation.

(4) The mean speed of the engine should be precisely that at which it is intended to be operated, irrespective of variation of load or of steam-pressure.

The Computation of Power, or of the work of the steam on the piston, is possible whenever, the dimensions of the engine being known, the mean pressure on the piston and its mean velocity have been measured for the required period. The mean effective pressure can only be obtained by the use of the indicator and from its diagram. The mean speed of piston is readily computed when, by counting the revolutions of the engine by the eye, watch in hand, or by the use of any convenient and reliable form of counter, the average number is obtained for the unit of time. The mean effective pressure

in pounds per square inch and the velocity of the piston in feet per minute being thus ascertained, the product of these factors into the *net* area of piston in square inches—the area of the rod being deducted on that side—gives the work, $2p_mLAN$, in foot-pounds per minute; and the indicated horse-power,

$$\text{I. H. P.} = \frac{2p_mLAN}{33000},$$

is at once given, A being the net area of piston.

If working up many diagrams from the same engine, the first step should be the computation of the "constant of the engine," a figure which expresses the horse-power of the engine, under its regular conditions of operation, for each pound mean pressure on the piston. Thus,

$$\text{I. H. P.} = \frac{2p_mLAN}{33000} = \frac{2p_mVA}{33000},$$

and when $p_m = 1$, this becomes

$$C = \frac{2LAN}{30000} = \frac{2VA}{33000},$$

where the symbols are those customarily used. Each diagram is then measured up and its mean pressure obtained, and the multiplication of this quantity by the constant thus computed gives the horse-power for that diagram.

69. The Steam or Water Consumption of the engine cannot be exactly ascertained by the use of the indicator, since a portion of the steam entering the engine is always instantly condensed by contact with the cooler walls of the cylinder, and another portion, sometimes considerable in amount, may escape past the piston, or through the valves, by leakage. The indicator does exhibit, however, the pressure and volume of steam actually present at each instant in the steam-cylinder, and it thus becomes easy to compute its weight and to obtain a measure of the quantity thus shown by the indicator, for comparison with the total quantity supplied by the boiler, and

thus to ascertain the losses, by condensation and leakage, of power, heat, and steam. The pressure being shown on the diagram at every point in the stroke, the "steam-tables" give the corresponding specific weights, the weights per unit of volume; and the space traversed by the piston up to the given point, plus the clearance-space, measures the actual volume: the latter quantity, multiplied by the specific weight, is the weight of uncondensed steam present in the cylinder at the specified point. The mean weight per stroke, multiplied by the number of strokes, being compared with the total weight supplied by the boiler in the same time, as shown by the log of the boiler-trial, the difference is the waste by internal condensation and leakage.

The real measure of the efficiency of any engine is the quantity of steam used by it to develop unity of power, and that efficiency is the greater the smaller its legitimate demand for steam, and the less its waste in these directions. Should it be impracticable to conduct a boiler-trial to determine the amount of steam drawn off to supply the engine, it may be possible to secure a fairly approximate measure—when it is known that the boiler gives dry steam—by observing the rate of fall of the water-level with the feed shut off and computing the volume and weight evaporated from the known dimensions of the volume thus traversed by the subsiding water-surface. Care must be taken not to allow its fall to go so far as to become a source of risk. It is usually easy to measure the volume corresponding to a fall nearly equal to the length of the gauge-glass. In many cases the quantity of steam shown by the indicator, at the point of cut-off, may be determined from the diagram, and a known or probably fair allowance may be added for unindicated wastes, to obtain an approximate measure of the quantity of water demanded per horse-power and per hour. This waste has been seen to be rarely as little as ten per cent., and often as much as thirty per cent. and upward.

The volume added by the clearance and port passages varies greatly with the type and build of the engine. In the single-valve and in the older forms of poppet-valve engines it

is rarely less than six, and is often ten, per cent. and more: in the best of modern engines it is often as low as two per cent. It may be easily measured, either from the drawings of the engine or by filling these spaces with water from a quantity previously weighed. The weight required to fill the clearances and ports gives the means of computing their volume from the known density of the liquid. Where, as is usual in recent and well-designed engines, considerable compression is employed, the saving of steam thus effected is to be carefully allowed for in all determinations of steam accounted for by the indicator. The loss by leakage should be inappreciable in any good engine, and this being ascertained by test, giving steam at one end and observing the escape of steam, if any occurs, by opening the indicator-cock on the other end, the whole waste shown will be due to cylinder condensation, the amount of which, as a percentage of the steam accounted for by the indicator, and as the quantity to be added to the latter for engines of fair size and good construction, may be taken as, approximately, from about $0.15 \sqrt{r}$ in the best cases of compound engines, to $0.2 \sqrt{r}$ in ordinary good unjacketed engines, and above the latter figure for engines of older type and slower speeds of rotation; r being the ratio of expansion for a single cylinder only, in the case of the compound engine, and that cylinder being taken which has the highest value of r.

When the problem to be solved is, as usual, the determination of the efficiency of any actual engine, as distinguished from the simple thermodynamic efficiency of the ideal engine, the indicator aids the engineer in its solution by showing the precise quantity of steam present at every instant during the stroke, and, hence, the quantity of water present at the same time; the sum of these two weights being always, if the piston and valves are tight, equal to the weight of feed-water passing through the boiler and entering the engine as a mixed working fluid. The volume and pressure of the steam are shown by the indicator, and the weight is easily computed from its known density at the given pressure. The portion of stroke traversed at any instant, added to the clearance-space,

measured in equivalent cylinder length, gives the volume of steam present. The quantity of steam supplied is equal to the measured quantity at the point of cut-off, less that retained by compression. The difference between the weight of steam thus measured at any point in the stroke, and the total weight of feed-water supplied, or steam entering the engine, per stroke, is the weight of water present.

Also, the total weight of mixed steam and water present from the point of cut-off to the opening of the exhaust-port is the sum of the quantity coming from the boiler and that compressed into the clearance-spaces. The variation of this quantity is well shown by the following data obtained by Mr. Spangler:*

Weight of steam	per I. H. P. per hour,		. . .	lbs.	28.15
" " priming	" "	9 per cent.,	. .	"	2.78
" " feed-water	" "		. . . .	"	30.93
" " steam at 0.9 stroke,			. . . .	"	20.06
" " water " "	"	33 per cent., nearly,		"	10.87
" " steam " 0.7	"		. . . .	"	19.23
" " water " "	"	38 per cent., nearly,		"	11.70
" " steam " 0.5	"		. . . .	"	18.27
" " water " "	"	40 per cent., nearly,		"	12.66
" " steam " 0.3	"		. . . .	"	17.31
" " water " "	"	58 per cent., nearly,		"	13.62

These figures, as will be seen by comparison with other similar data, are indicative of a greater waste than usually occurs in large engines, due to the small size of that here referred to. It is evident that all variations from the proportions of the mixture entering the engine must be due to the transfer of heat to and from the metal of the cylinder and piston. The above figures show a gradual increase of the proportion of steam present produced by re-evaporation of that condensed initially, on entrance, from the point of cut-off up to the end of the stroke.

The weight of steam in the cylinder at any point of the

* Journal Franklin Institute, Feb. 1886.

stroke, in pounds per indicated horse-power per hour, is always equal to

$$\frac{W}{\text{I. H. P.}} = \frac{60 \times 2lanw'}{144} \div \frac{2p_m lan}{33000} = \frac{13750w'}{p_m} = w,$$

or

$$w = \frac{a}{p_m}\left[\left(\frac{1}{r} - \frac{c}{v_1}\right)w' - \frac{(c'-c)w''}{v_1}\right];$$

in which

p_m = mean effective pressure;

c and c' = the volume of clearance-space and of steam at the point of closing of the exhaust-valve and beginning of compression;

$\frac{c}{v_1}$ and $\frac{c'}{v_1}$ = their ratios to total cylinder volume;

r = ratio of the given travel to full stroke of piston;

w, w', and w'' = weight of steam per indicated horse-power per hour; the specific weight at the pressure found at the given point; and the specific weight at the beginning of compression;

a = when the piston area is measured, as above, in square inches and the stroke in feet,

$$\frac{1980000}{144} = 13750.$$

This computation is commonly made for the points of cut-off and of release. At the former the "initial condensation" is obtained, and probably the best measure of the waste by condensation; at the latter a measure is secured of the state of the mixture exhausted from the engine.

The following example, from a diagram taken by Barrus,* employing Clarke's tables,† illustrates these computations: Taking $\frac{1}{r} = 0.308$ at cut-off, and $\frac{1}{r} = 0.901$ at release, $c = 0.02v_1$;

* On the Tabor Indicator, p. 48.

† Manual for Mechanical Engineers.

$c' = 0.071v_1$; $p_m = 38.45$; $w' = 0.1844$ at cut-off and 0.0705 at release; $w'' = 0.0457$. Then, at cut-off,

$$w = (13750 \div 38.45)[(0.308 + 0.02)0.1844 - (0.071 + 0.02)0.0457] = 20.13 \text{ lbs.};$$

and, at release,

$$w = (13750 \div 38.45)[(0.91 + 0.02)0.0705 - (0.071 + 0.02)0.0457] = 21.95 \text{ lbs.}$$

Here, the indicator, in the instance from which these figures are derived, shows a difference in computed weights of steam per horse-power and per hour at the points of cut-off and release amounting to nearly two pounds—about ten per cent., which difference is the measure of the re-evaporation taking place during expansion. To the figures above obtained must be added the allowances for total wastes.

We have in many instances so little compression that it may be neglected. In such cases the following very simple process may be adopted: Assuming the working fluid to be water instead of steam, the quantity demanded would be, per horse-power per hour,

$$w_a = \frac{1980000 \times 62.4}{144} = 857900$$

pounds at one pound pressure per square inch; and, at any other pressure, p,

$$w_b = \frac{857900}{p};$$

while if steam be employed the weight would be less in proportion to its greater, its specific, volume, v', and the weight actually needed would be

$$w_c = \frac{857900}{p_m v'}, \text{ nearly};$$

which may be corrected for clearance and compression.

The detailed method of computation from this indicator-diagram is illustrated fully, later (§ 71).

The following is a convenient form of this expression for steam and water consumption:

Let p_1 = initial pressure, absolute;
p_2 = back-pressure, absolute;
r = true expansion-ratio;
c = clearance fraction;
D = density of steam in lbs. per cu. ft.;
w = weight of steam per horse-power per hour;

$$w = \frac{a}{p_1} \cdot \frac{1 + c\left(1 + \frac{p_2 r}{p_1}\right)}{(1 + \log_e r)(1 + c) - \left(\frac{p_2}{p_1} + c\right)r}.$$

The constant has been seen to have the value, in British measures, $a = 13750$. Compression is here neglected. This expression assumes a minimum value, for the ideal case, as is elsewhere seen, where

$$r = \frac{1 + c}{\frac{p_2}{p_1} + c}, \text{ nearly.}$$

This value is, in the real engine, found to be greatly reduced by the occurrence of internal wastes, by cylinder condensation or leakage.

The following table, prepared by Mr. Thompson, gives the factors employed in computing the indicated consumption of water.* The method is illustrated in Fig. 97. The mean effective pressure must be known, but the horse-power or the size of the cylinder need not be known. Draw a vertical at each end of the diagram, and continue the expansion-curve to t. From t draw tC. Measure the absolute pressure at t, and find in the table, page 244, the corresponding number. Numbers

* Hemenway's Indicator Practice; N. Y., 1886; J. Wiley & Sons. American Machinist.

WATER-COMPUTATION TABLE.

T P	0	1	2	3	4	5	6	7	8	9
3	117.300	121 015	124.717	128.406	132.083	135.748	139.399	143 075	146 665	150 279
4	153.880	157.514	161.137	164 750	168.353	171 945	175.527	179 098	182 659	186 210
5	189 750	193 336	196 914	200.483	204 044	207 598	211.142	214 679	218.208	221 728
6	225 240	228 799	232 351	235 897	239 437	242.970	246 497	250.017	253.531	257.039
7	260 540	264 056	267.566	271 071	274 570	278.063	281.550	285 031	288.506	291 976
8	295 440	298 922	302 400	305 872	309 338	312 800	316.256	319 708	323.154	326 594
9	330 030	333 483	336 941	340 389	343.833	347 273	350.707	354.137	357 563	360 984
10	364 400	367 842	371 280	374 714	378 144	381.570	384.992	388.410	391 824	395 234
11	398 640	402.064	405 485	408 902	412 315	415.725	419 131	422.534	425 933	429 328
12	432 720	436 120	439 517	442 911	446.301	449 688	453 071	456.451	459.828	463 200
13	466 570	469 950	473 326	476.699	480 068	483.435	486 798	490 159	493.516	496 869
14	500 220	503 596	506 968	510.338	513.706	517.070	520 432	523.790	527.146	530 500
15	533.850	537 213	540 573	543.930	547.285	550.638	553 987	557 334	560 679	564 011
16	567 300	570 713	574 063	577 411	580.757	584 100	587 441	590 780	594.115	597 449
17	600.780	604 109	607 435	610.759	614.081	617 400	620 717	624 031	627.343	630 653
18	633 960	637 265	640 567	643.867	647 165	650 460	653 753	657 043	660.331	663 617
19	666 900	670 200	673 498	676 793	680.086	683.378	686 666	689 953	693.238	696 520
20	699 800	703 098	706 394	709 688	712 980	716 270	719 558	722 844	726 128	729 410
21	732 690	735 968	739.244	742 518	745.790	749.060	752.328	755 594	758.858	762 120
22	765.380	768 660	771.938	775 215	778.490	781 763	785 034	788.303	791 570	794 836
23	798 100	801.362	804.622	807 881	811.138	814.393	817 646	820.897	824.146	827 334
24	830 640	833 908	837 175	840 440	843.703	846 965	850.225	853.484	856 741	859.996
25	863 250	866 502	869 753	873.002	876.249	879.495	882.739	885.982	889 223	892 462
26	895.700	898 936	902 171	905 404	908 635	911 865	915 093	918 320	921 545	924.768
27	927 990	931 210	934 429	937 646	940 831	944 075	947 287	950.498	953.707	956 914
28	960 120	963 352	966 583	969.813	973 041	976.268	979 493	982 717	985 939	989 160
29	992 380	995 598	998.815	1002 031	1005 245	1008 458	1011 669	1014 879	1018.087	1021 294
30	1024 500	1027 704	1030 907	1034.109	1037 309	1040 508	1043 705	1046 901	1050.095	1053 288
31	1056 480	1059 670	1062 859	1066 047	1069 233	1072.418	1075 601	1078 783	1081 963	1085 142

WATER-COMPUTATION TABLE—*Continued.*

T. P.	0	1	2	3	4	5	6	7	8	9
32	1088.320	1091.528	1094.736	1097.942	1101.146	1104.350	1107.552	1110.754	1113.954	1117.152
33	1120.350	1123.546	1126.742	1129.936	1133.128	1136.420	1139.510	1142.700	1145.888	1149.074
34	1152.260	1155.444	1158.628	1161.810	1164.990	1168.170	1171.348	1174.526	1177.702	1180 876
35	1184.050	1187.222	1190.394	1193.564	1196.732	1199.900	1203.066	1206.232	1209.396	1212.558
36	1215.720	1218.917	1222.112	1225.307	1228.500	1231.693	1234.884	1238.075	1241.264	1244.453
37	1247.640	1250.827	1254.012	1257.197	1260.380	1263 563	1266.744	1269.925	1273.104	1276.283
38	1279 460	1282.637	1285.812	1288.987	1292.160	1295.333	1298.504	1301.675	1304.844	1308.013
39	1311.180	1314.347	1317.512	1320.677	1323.840	1327.003	1330.164	1333.325	1336.484	1339 643
40	1342.800	1345.957	1349.112	1352.267	1355 420	1358.573	1371.724	1364.875	1368.024	1371.173
41	1374.320	1377.467	1380.612	1383.757	1386.900	1390.043	1393.184	1396.325	1399.464	1402.603
42	1405.740	1408.877	1412.012	1415.147	1418.280	1421.413	1424.544	1427.675	1430.804	1433.933
43	1437.060	1440.230	1443.398	1446.566	1449.734	1452.900	1456.066	1459.230	1462.394	1465.558
44	1468.720	1471.882	1475.042	1478.202	1481.362	1484.520	1487.678	1490.834	1493.990	1497.146
45	1500.300	1503.454	1506.606	1509.758	1512.910	1516.060	1519.210	1522.359	1525.506	1528.654
46	1531.800	1534 946	1538.090	1541.234	1544.378	1547.520	1550.662	1553.802	1556.942	1560.082
47	1563.220	1566.358	1569 494	1572.630	1575.766	1578.900	1582.034	1585.166	1588.298	1591.430
48	1594.560	1597 690	1600.818	1603.946	1607.074	1610.200	1613.326	1616.450	1619.574	1622.698
49	1625.820	1628.942	1632.062	1635.182	1638.302	1641.420	1644.538	1647.654	1650.770	1653.886
50	1657.000	1660.114	1663.226	1666.338	1669.450	1672.560	1675.670	1678.778	1681.886	1684.994
51	1688.100	1691.206	1694.310	1697.414	1700.518	1703.620	1706.722	1709.822	1712.922	1716.022
52	1719.120	1722.218	1725.314	1728.410	1731.506	1734.600	1737.604	1740.786	1743.878	1746.970
53	1750.060	1753.150	1756.238	1759.327	1762.414	1765.500	1768.586	1771.670	1774.754	1777.838
54	1780.920	1784.002	1787.082	1790.162	1793.242	1796.320	1799.398	1802.474	1805.550	1808.626
55	1811.700	1814.829	1817.957	1821.084	1824.211	1827.338	1830 463	1833 588	1836.713	1839.837
56	1842.960	1846.083	1849 205	1852.326	1855.417	1858.568	1861.687	1864.806	1867.925	1871.043
57	1874.160	1877.277	1880.393	1883.508	1886.623	1889.738	1892.851	1895.964	1899.077	1902.189
58	1905.300	1908.411	1911.521	1914 630	1917.739	1920.848	1923.955	1927.062	1930.169	1933.275
59	1936.380	1939.485	1942.589	1945.692	1948.795	1951.898	1954.999	1958.100	1961.201	1964.301
60	1967.400	1970.499	1973.597	1976.694	1979.791	1982.888	1985.983	1989.078	1992.173	1995.267

under T. P. represent the terminal pressure in pounds, and the figures, 1, 2, 3, etc., tenths of a pound.

Divide this number by the mean effective pressure; the quotient will be the steam accounted for per horse-power per

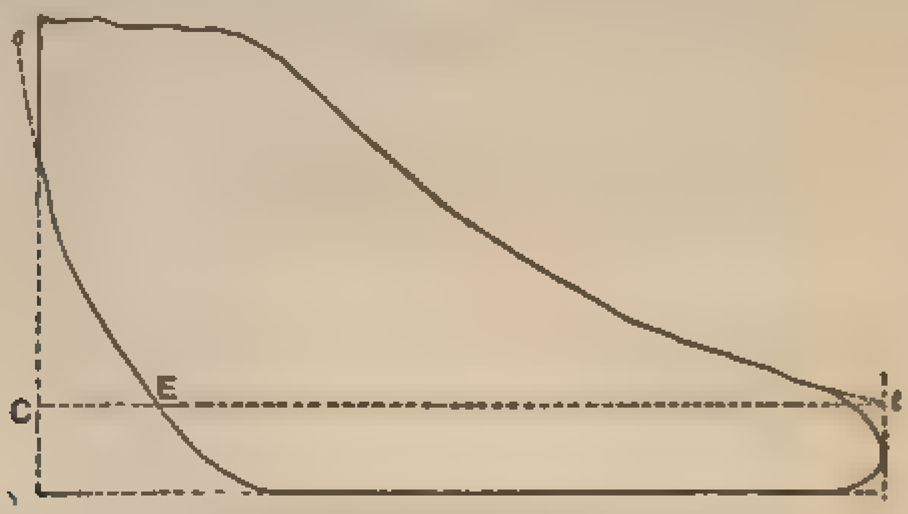

FIG 97.—INDICATED STEAM VOLUMES. SCALE 40.

hour, uncorrected for compression. To make this correction multiply by *tE* and divide by *tC*.

When the maximum compression is not as high as the terminal pressure, the compression-curve must be extended, as *c*, and *E* will be outside the diagram.

In illustration of the computation of the economical performance of engines by determining their expenditure of heat, measured in British thermal units, may be given the following, as computed by Mr. Barrus,* using Clark's tables.†

Assume a non-condensing engine to use 27 lbs. of feed-water per horse-power per hour, supplied at 212° F.; a condensing engine 18 lbs., at 130°; and a compound engine 14 lbs., at 170°. The pressure in the first two cases is 80 lbs. and in the last case 120. In the first two cases the steam contains ½ per cent moisture, and in the last case it is superheated 20°. The total heat of saturated steam of 80 lbs. pressure [94.7 lbs. absolute] is 1212.2 *B. T. U.* Deduct the heat corresponding to 0.05 moisture, $0.05 \times 885.9 = 4.4$ [885.9 being the latent heat], and there remain 1207.8 units, the total heat of steam

* The Tabor Indicator.

† Manual for Mechanical Engineers.

containing ½ of one per cent of moisture, measured above zero. Deduct the heat corresponding to a feed-water temperature of 212°, 212.9 thermal units, and there remain 994.9 units, the total heat of one pound of steam, containing ½ of one per cent moisture, above the temperature of feed-water. Multiply this by 27, and the product, 26,862.3 units, is the heat expended per horse-power per hour.

A similar computation gives 19,400.4 thermal units per horse-power per hour for the second case.

In the third case, the total heat of saturated steam of 120 lbs. pressure [134.7 lbs. above zero] is 1220.1 *B. T. U.* The heat corresponding to 20° superheating is $20 \times 0.475 = 9.5$, which gives 1229.6 units for the total heat of superheated steam. Deduct 170.4 thermal units, the heat corresponding to a feed-water temperature of 170°, and multiply by 14, and we have for a product 14,828.8 units, the heat expended per horse-power per hour.

These results are tabulated below:

Kind of Engine.	Non-Condensing. *A.*	Condensing. *B.*	Compound. *C.*
Boiler-pressure ....lbs.	80	80	120
Av. temp. of feed-water...............deg.	212	130	170
Feed-water per H. P. per hour......... lbs.	27	18	14
Percentage moisture in steam......... %	0.5	0.5	20° superh.
Total heat of saturated steam........th. un.	1212.2	1212.2	1220.1
Total heat corrected for moisture and superheating..........................th. un.	1207.8	1207.8	1229.6
Heat of feed-water... .... "	212.9	130	170.4
Heat expended per pound............ "	994.9	1077.8	1059.2
Heat expended per H. P. per hour.... "	26.862.3	19,400.4	14,828.8

A comparison of the heat thus computed, as expended, with the heat-equivalent of the useful work performed, determines the efficiency.

As each horse-power is the thermal equivalent of 42.75 heat-units per minute or 2565 units per hour, we have for the three cases,

$$E: \quad \overset{A.}{\frac{2565}{26862}} = 0.096; \qquad \overset{B.}{\frac{2565}{19400}} = 0.137; \qquad \overset{C.}{\frac{2565}{14829}} = 0.16;$$

or efficiencies of 9.6, 13.7, and 16 per cent., as compared with an engine of efficiency unity, perfectly utilizing all the heat-energy supplied to it. This is the method first adopted by Rankine, except that thermal, rather than mechanical, units are employed.

70. Constructing Hyperbolic Curves, such as are commonly taken to represent the variations of pressure and volume in the ideal diagram, enables the engineer to obtain some idea of the method and extent of variation of the actual quantities in real engines from those of the ideal case. There are several methods of constructing these curves, of which the simplest are, perhaps, the following, as applied to produce the equilateral hyperbola, the curve of Mariotte, to which the expansion-line, in the best classes of engine, very closely approximates, and which is commonly taken as the standard.

Let XX, YY be given asymptotes (i.e., the clearance and true vacuum lines of the indicator-card) and x any given point, and let yx, xy be its co-ordinates.

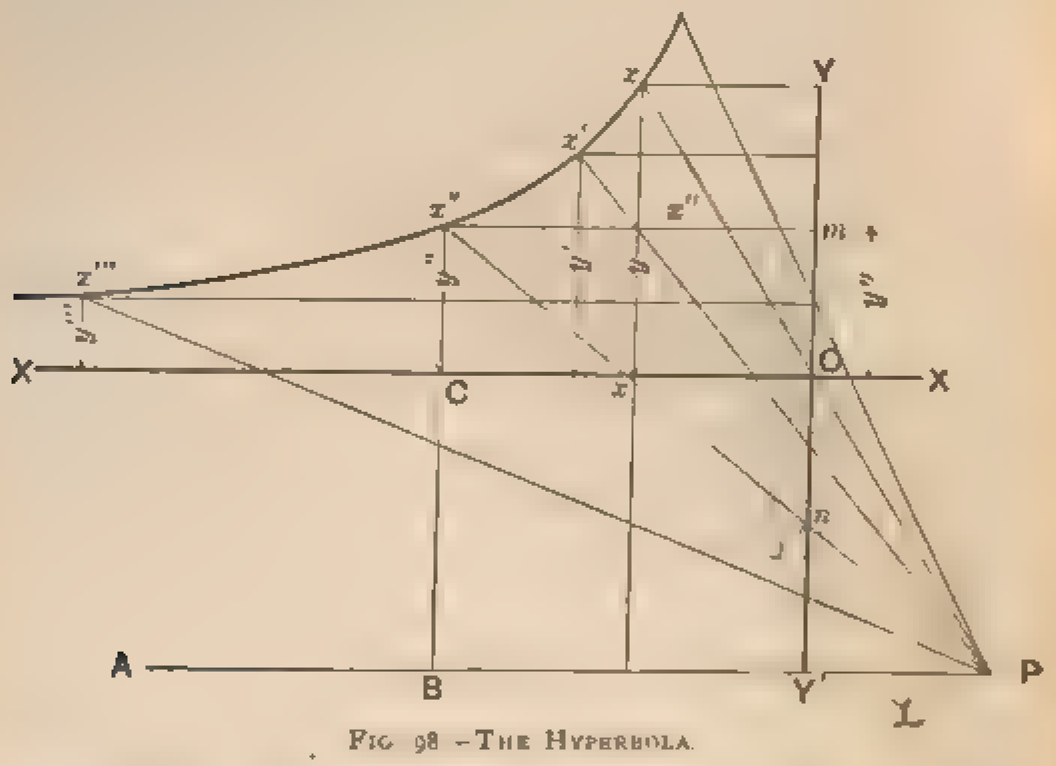

FIG. 98.—THE HYPERBOLA.

Extend YO until $OY' = YO$ and draw AP, making $Y'P$ equal to xY and parallel to XX.

Divide YO and OY' into similar divisions.

Assume an ordinate Om of a point to be found, and draw mx'' parallel to XX.

At Y' erect $Y'n = Om$, and draw Pnx''; the point x'' of intersection with $x''n$ is the required point.

For in the triangles $nY'P$, nmx'' we shall have

$$nY' : Y'P :: mn : x''m = \frac{xy}{y''} = x'':$$

$$\text{i.e., } y'' : x :: y : x''. \quad \text{Q. E. D.}$$

When the expansion-line is true to the hyperbolic curve, it becomes possible to obtain a fairly approximate measure from the diagram of the clearance-space; or, the latter being known, to determine the real locus of the hyperbolic expansion-curve, as follows:

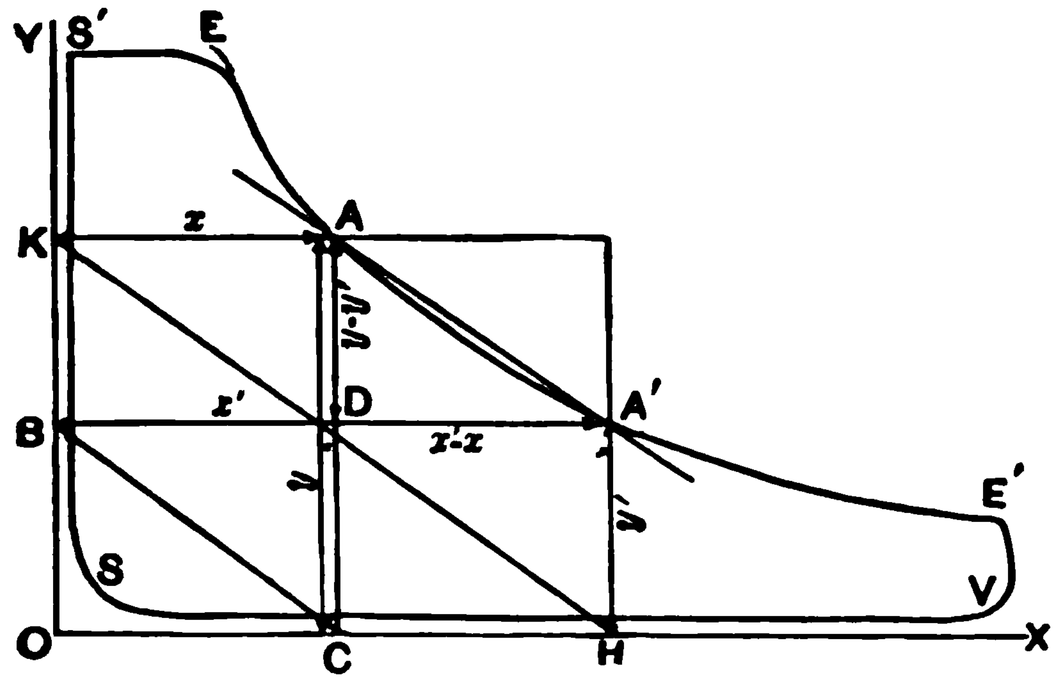

FIG. 99.—THE HYPERBOLIC EXPANSION-LINE.

Let S', E, E', V, S represent an indicator-card; let OX be the line of perfect vacuum; OY the line at end of cylinder plus the clearance; then, OX and OY will be asymptotes of the hyperbola E, A, A', E', the curve of expansion.

Take two points on the curve AA', and AK, AC, $A'B$, and $A'H$ will be their co-ordinates.

Draw AA', and from C, the line CB parallel to AA'; the point B, where it intersects $A'B$, will be a point in the line OY.

Or, draw HK parallel to AA', and K, the intersection with AK, will be such a point.

For by Mariotte's law and from the properties of the hyperbola $xy = m$; $x'y' = m$; $\therefore xy = x'y$.

$$\therefore x : x' :: y' : y;\ x' - x : x :: y - y' : y';$$
$$\text{or, } A'D : BD :: AD : DC.$$

And, from similar triangles (by construction),

$$A'D : BD :: AD : DC'. \quad \text{Q. E. D.}$$

Conversely, having given the clearance and the scale of the indicator, with point of cut-off, *to find the expansion-line.*

In proportion $y - y' : y' :: x' - x : x$, assume x' and find values of y' by constructing the triangle KGH, similar to ADA'.

Taking the point of release as a point in the hyperbolic curve, and laying down that curve on the diagram, it will be found, not only that the curve and the expansion-line of the diagram do not coincide, but that the latter falls above the former throughout its length, in nearly all cases, indicating, usually, initial condensation and later re-evaporation, but sometimes indicating some leakage as well. If the weight of steam actually drawn from the boiler be taken as the basis of a diagram, using its volume as the initial ordinate of the hyperbolic curve, it becomes easy to trace the variations of the whole actual diagram from the ideal indicator-card, as here shown.

In any case in which the curve represented by the expansion-line is of the class of which the equation is

$$pv^n = p_1v_1^n = p_2v_2^n,$$

the co-ordinates sought, any one point, p_1v_1 or p_2v_2 being given, may be found, and any new point in the ideal curve determined by computation, thus: From the above expression,

$$n \log v + \log p = n \log v_1 + \log p_1;$$

and if p_1 and v_1 are known, for any assumed volume v, the logarithm of the corresponding new pressure must be

$$\log p = n \log v_1 + \log p_1 - n \log v;$$

which expression being used to determine several points, the curve may be drawn through them.

The values of n have been seen to be as follow:

Equilateral hyberbola, ...	1
Curve of steam; saturation $\frac{17}{16}$, or	1.0646
Adiabatic curve, steam, ...	$1.035 + 0.1x$
" " gas, ...	1.408
Isothermal " "	1.0

The variation of the actual ratios of expansion from their apparent values, in engines having large clearance-spaces, is very considerable at high ratios of expansion and in short-stroke engines. The following table (p. 252) published by Mr. Grimshaw, is sufficiently extensive for ordinary purposes, and well exhibits those differences.*

The close approximation of the three principal steam-ex-

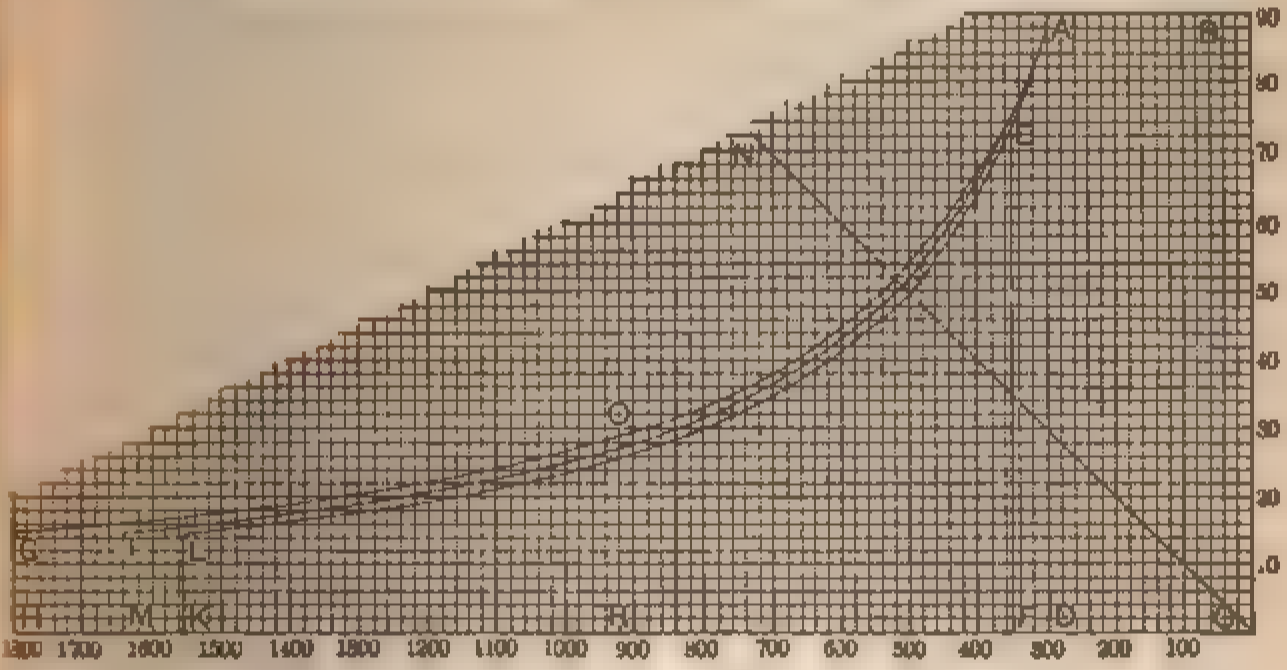

FIG. 100.—THE THREE EXPANSION CURVES.

pansion lines is well shown by the accompanying diagram, a set of curves shown in various publications, but probably first laid down in this form by Mr. Porter.† AB exhibits the initial volume, as does also CD; AD and BC represent the initial pressure; EF is an ordinate, taken at convenience; and the terminal ordinates are GH, IM, and LK. OR is taken at half-stroke; while CN is the axis of the equilateral hyper-

* Am. Machinist, Jan. 20, 1883, p. 5.

† Steam-engine Indicator, p. 123.

REAL RATIOS OF EXPANSION.

Per cent. of Clearance.	Points of Cut-off.															
	10	.125	20	25	.30	333	.375	40	50	60	625	.70	75	.80	.875	.90
.01	9.111	7.481	4.809	3.884	3.258	2.944	2.623	2.463	1.983	1.655	1.590	1.422	1.328	1.246	1.141	1.109
.0125	9	7.363	4.764	3.875	3.24	2.930	2.612	2.454	1.975	1.653	1.588	1.421	1.327	1.246	1.140	1.109
.0150	8.826	7.25	4.720	3.830	3.222	2.9 6	2.602	2.445	1.970	1.650	1.585	1.419	1.326	1.245	1.140	1.109
.0175	8.659	7.133	4.677	3.803	3.204	2.902	2.592	2.436	1.966	1.647	1.583	1.418	1.325	1.244	1.140	1.108
.02	8.5	7.034	4.635	3.777	3.187	2.889	2.582	2.428	1.961	1.645	1.581	1.416	1.325	1.243	1.138	1.108
.0225	8.346	6.932	4.595	3.752	3.170	2.876	2.574	2.420	1.956	1.642	1.579	1.415	1.324	1.243	1.138	1.108
.0250	8.2	6.833	4.555	3.727	3.153	2.863	2.562	2.411	1.952	1.640	1.576	1.413	1.322	1.242	1.138	1.108
.0275	8.088	6.738	4.516	3.702	3.137	2.850	2.552	2.403	1.947	1.637	1.574	1.412	1.321	1.241	1.138	1.107
.03	7.933	6.645	4.417	3.678	3.121	2.837	2.543	2.395	1.943	1.634	1.572	1.410	1.320	1.240	1.138	1.107
.0325	7.742	6.555	4.440	3.654	3.105	2.824	2.533	2.387	1.938	1.632	1.570	1.409	1.319	1.240	1.138	1.107
.0350	7.660	6.468	4.404	3.631	3.089	2.812	2.524	2.379	1.934	1.629	1.568	1.408	1.318	1.239	1.137	1.106
.0375	7.545	6.390	4.484	3.608	3.074	2.800	2.515	2.371	1.930	1.627	1.506	1.406	1.317	1.238	1.136	1.106
.04	7.428	6.303	4.333	3.58	3.058	2.788	2.506	2.363	1.925	1.625	1.563	1.405	1.316	1.238	1.136	1.106
.0425	7.315	6.229	4.298	3.564	3.043	2.776	2.497	2.355	1.921	1.622	1.561	1.404	1.315	1.237	1.136	1.106
.0450	7.206	6.147	4.256	3.542	3.028	2.764	2.488	2.348	1.917	1.620	1.569	1.402	1.314	1.236	1.135	1.105
.0475	7.102	6.082	4.232	3.521	3.014	2.752	2.479	2.340	1.913	1.617	1.557	1.401	1.313	1.235	1.135	1.105
.05	7	6	4.2	3.5	3	2.741	2.470	2.333	1.907	1.615	1.555	1.400	1.312	1.235	1.135	1.105
.0525	6.901	5.985	4.168	3.478	2.986	2.730	2.461	2.325	1.904	1.613	1.553	1.398	1.311	1.234	1.134	1.104
.0550	6.806	5.861	4.130	3.459	2.971	2.719	2.453	2.318	1.900	1.610	1.551	1.397	1.310	1.233	1.134	1.104
.0575	6.714	5.794	4.106	3.439	2.957	2.708	2.445	2.311	1.896	1.608	1.549	1.396	1.309	1.233	1.134	1.104
.06	6.625	5.729	4.076	3.418	2.944	2.697	2.436	2.304	1.892	1.606	1.547	1.394	1.308	1.232	1.133	1.104
.0625	6.538	5.666	4.047	3.407	2.931	2.686	2.428	2.297	1.888	1.603	1.545	1.393	1.307	1.231	1.133	1.103
.0650	6.454	5.605	4.045	3.380	2.917	2.675	2.420	2.290	1.884	1.601	1.343	1.392	1.306	1.231	1.132	1.103
.0675	6.373	5.545	3.990	3.362	2.904	2.665	2.412	2.283	1.881	1.599	1.541	1.390	1.305	1.230	1.132	1.103
.07	6.294	5.482	3.963	3.342	2.892	5.655	2.404	2.276	1.877	1.597	1.539	1.389	1.304	1.229	1.132	1.103

bola, *AOG*, the upper curve, of which *CB* and *CH* are asymptotes. Ordinates measure absolute pressures in pounds per square inch; abscissas represent volumes of unity of weight (1 lb.). Thus *BA* is the volume (4.73 cu. ft.) of one pound of steam at a total pressure of 90 pounds per square inch; *ABCD* is the external work done in its production. It is this curve which is commonly assumed to be that of the expansion of steam.

The curve *AOI* is the curve of dry and saturated steam, its co-ordinates representing the simultaneous pressure and volume of the fluid when in contact with the mass of water from which it is produced. The expansion is less, and the rate of fall of pressure greater, than if it were to follow the law of Mariotte. It is this curve which is assumed to be described when steam expands in well-jacketed engines.

The lower line, *AOL*, is the adiabatic curve, assumed to be obtainable in engines with non-conducting cylinders and approximately in "high-speed engines." The area under this, as under the other curves, represents the work done as the steam expands, and exhibits the gain obtainable by expansion, in each case. In all real engines, however, the expansion-line falls at first more rapidly, and finally more slowly, than either of these curves. As elsewhere seen, this variation from the ideal curve is often very observable.

71. Cylinder Condensation and Leakage produce variations in the diagram, as obtained, which differently affect the different parts of the curve. Leakage can usually be eliminated, and always should be before the engine is set at work regularly. The first-named waste is usually irremediable. When the exact measure of the quantity of steam expended is obtained by a boiler-trial, it is easy to trace these variations, as in the diagram here given, as taken from the engine and worked up by the late Professor C. A. Smith, in which illustration the diagram which should have been produced by the same steam, had there been no initial condensation, is shown with the real diagram.*

* Steam making, p. 91

This indicator-diagram is an unusually good sample, as to form, and was taken from the St. Louis high-service pumping-engine, a machine of 705 I. H. P., 85 inches diameter of cylinder, and 10 feet stroke of piston, making 11½ revolutions per minute. Taking measures of the abscissas of the two diagrams, it is seen that the condensation amounts to from about 30 per cent. as a minimum to 50 per cent. as a maximum, so far as measurable, the actual card illustrating the expansion in a metallic cylinder of the steam, which would have given the larger diagram in an ideal engine with its non conducting cylinder. The complete ideal diagram would extend proportionally farther toward the right and beyond the limits of the actual figure. When the two lines continue so far separated,

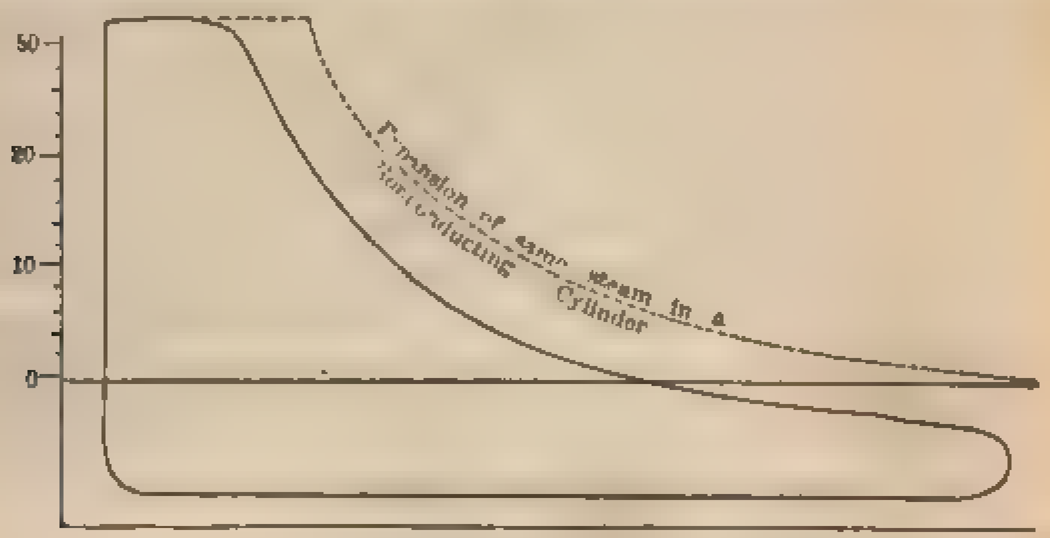

FIG. 101.—THE REAL AND THE IDEAL CARD.

it is an indication of large initial condensation, and correspondingly great re-evaporation after the exhaust-valve opens; as the initial condensation is due to, and is proportional to, the re-evaporation. In most cases, however, the engineer, unable to determine these data, assumes the point of release, or the point of intersection of the expansion-line prolonged with the ordinate at the extreme end of the diagram, as that of coincidence of the ideal and the real curve, and draws the hyperbolic curve backward from that as a given point, in the manner already described. A comparison of the ideal diagram thus formed with the actual indicator-card will give a means of judging of the character of the engine studied as a thermo-

dynamic apparatus, and of comparing different engines. An exact coincidence of the two diagrams, in any given case, would not prove, or even give a presumption of freedom from such waste; nor would the equality, in this respect, of diagrams from any two engines prove more than a probable general similarity in their performance, thermodynamically. Such comparisons are, nevertheless, both interesting and instructive, as is seen in the following examples. They also give some indications of the probable consumption of water and steam, the real gauge of the efficiency of engines. The clearance may be determined by measurement of the engine, by the graphical method described in the preceding article, or by the following simple methods of construction.*

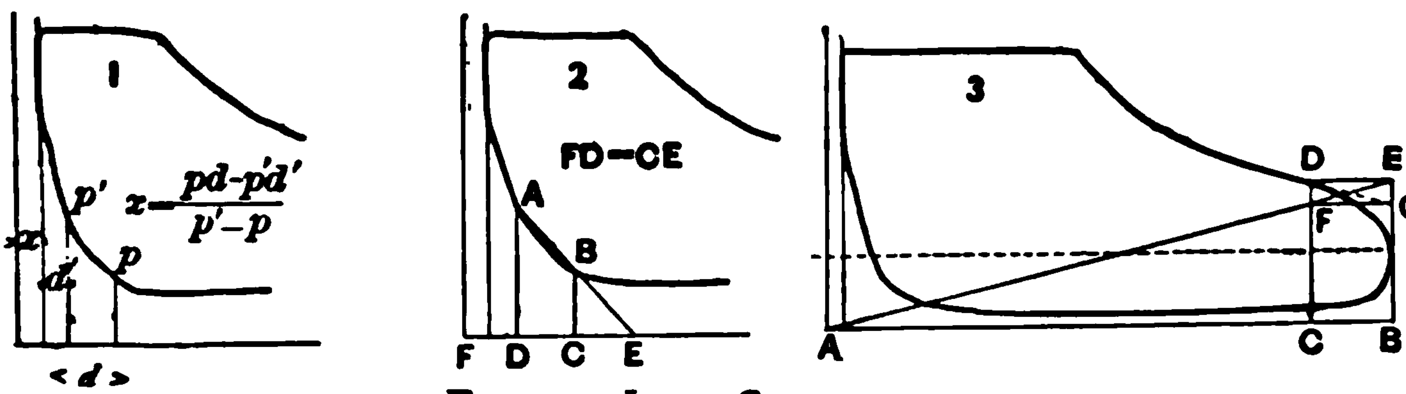

FIG. 102.—IDEAL CONSTRUCTIONS.

In case 1 let p and d, p' and d', be co-ordinates of the two given points, and x = the clearance; then

$$(x+d)p=(x+d')p', \qquad \text{and} \qquad x=\frac{pd-p'd'}{p'-p}.$$

Or we may determine the clearance geometrically by the following construction (see case 2). Assume two points A and B in the compression-curve; connect them by a right line, AB, continuing this line until it cuts FE at E. Draw AD and BC perpendicular to FE, and make $FD = CE$. Then F is the end of the ideal diagram including clearance, and the distance of F from the end of the indicator-diagram is the clearance.

To lay out the theoretical diagram: Draw a line representing the boiler-pressure and also a line of perfect vacuum, at 14.7 pounds below the atmospheric line, unless the true baro-

* First published by Mr. G. H. Babcock; Journal Franklin Institute, Sept. 1869.

...ide the length of the diagram ... any number of equal parts, as ... the point of release, and find the ... ment method as that shown in ... length of the diagram, including ... release. Draw DE parallel to ... at F. Draw FG parallel to AB, ... terminal pressure, the tension at ... equal to the whole capacity of the

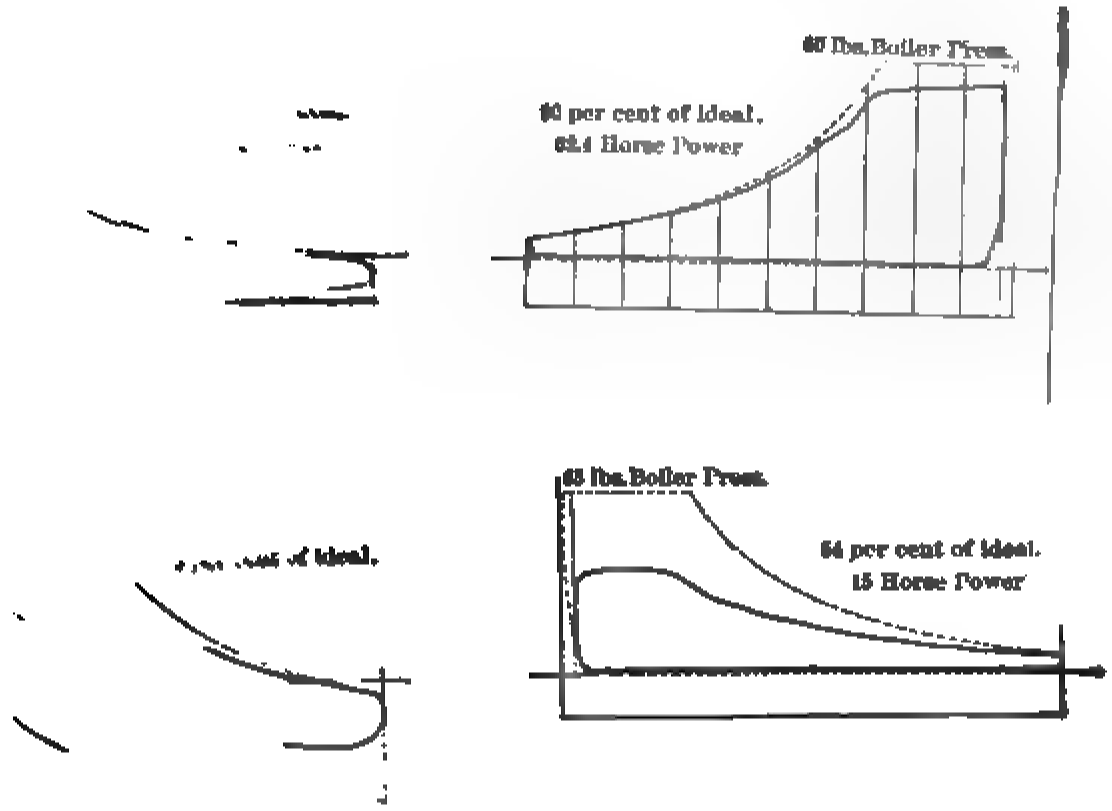

FIG. ...—IDEAL AND REAL DIAGRAMS.

... stance would be discharged at the termination ...

... sure at any other point of the stroke is easily found ... methods. With ten divisions, the several ordi... expansion-curve may be obtained by multiplying ... pressure by the following factors: 1, 1.11, 1.25, ... 2.5, 3.333, 5, 10. Having found the ideal ... each division, we trace a curve through these points ... the ideal point of cut-off, giving the same ter... as is observed in the actual case.

If the exhaust-valve closes before the end of the return stroke, so much of the cylinder full of steam as is thus imprisoned must be allowed for in the ideal diagram. Draw a hyperbolic curve tangent to the actual compression-line, and extending to the line of boiler-pressure, and thus find the boundary of the ideal diagram.

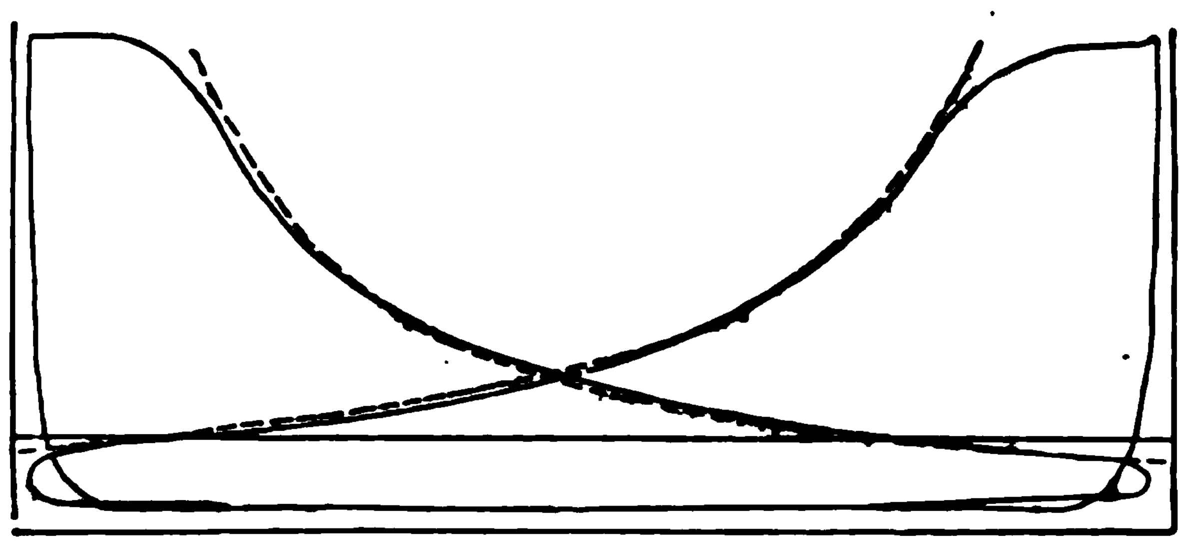

FIG. 104.—IDEAL AND REAL CARDS.

The group of four diagrams, Fig. 103, is given by Mr. Babcock in illustration of this method. The upper pair show a remarkable approximation of the actual to the standard figure, each giving, from the measured steam, 90 per cent. of the power which an engine having a non-conducting cylinder should give. One is a condensing, the other a non-condensing, mill engine; both designed by Mr. Babcock. The other pair are similar in their wastefulness, each giving but about one half the maximum, ideal, amount of work. One is from an old naval condensing engine, the other from a non-condensing stationary engine.

The next figure is a *fac-simile* of a pair of diagrams from an engine designed by Mr. J. W. Thompson, as studied by Mr. Hill, who gives the following analysis, using the curve of dry and saturated steam, having the equation $pv^{\frac{17}{16}} =$ constant as the standard. The engine was 22 inches diameter of cylinder, and 44 inches stroke of piston, making 70 revolutions per minute. The clearance is stated at .0175 piston-displacement.

The diagrams were measured with an "Amsler planimeter," and read as follows:

metric reading ... ...mos-
gram, including ... .19.9765 lbs.
ten. Measure ... ...mos-
terminal pr... .....10.143 lbs.
case 3, in wh...
clearanc... ....30.119 lbs. per sq. inch.
AB an... ...n, "constant" $\times p_m$,
and *Bc*...
whi... ...5 = 118.063 H. P.

...43 = 59.946 H. P.

...00 H. P.
... atmosphere to power above atmos-

... = 50.774 per centum.

... including cushion reads 31.134 lbs., and ... becomes

$$\frac{...34 - 30.119}{31.134} = .0326;$$

... 90.74 per centum of total capacity utilized.
... steam to produce the power according ... estimated as follows:

$$\ldots \times 70 \times 2 \times 60 = 81305.532 \text{ cu. ft.}$$

... displacement per hour.
... the diagrams (both ends of cylinder) appears ... inches from beginning of stroke, hence

$$\ldots \times \frac{43.175}{44} = 79781.05 \text{ cu. ft. to release.}$$

The exhaust closes (both ends of the cylinder) at 4.1712 inches from end of stroke (return); hence

$$\frac{81305.532 \times 4.1712}{44} = 7707.764 \text{ cu. ft.}$$

Clearance-volume $81305.532 \times .0175 = 1422.846$ cu. ft.

The volume of steam accounted for to release becomes

$$79781.05 + 1422.846 = 81203.896 \text{ cu. ft.,}$$

and the volume of steam retained in the cylinder by closure of exhaust becomes

$$7707.764 + 1422.846 = 9130.61 \text{ cu. ft.}$$

The terminal pressure is $\frac{11.5 + 12.75}{2} = 12.125$ lbs.,

and the weight of a cubic foot of steam at this pressure is obtained by Tate's formula, thus: 12.125 lbs. $= f = 24.7$ inches mercury; and a cubic foot of water at maximum density weighs, according to Berzelius, 62.388 lbs.; hence

$$\frac{62.388}{25.62 + \frac{49513}{f + .72}} = .0316 \text{ lbs.,}$$

and $81203.896 \times .0316 = 2566.043$ lbs. steam.

The steam retained by cushioning is as follows: The pressure in front of piston at time exhaust closes (both ends of cylinder) is 3.75 lbs., and the weight per cubic foot of steam at this pressure is

$$\frac{62.388}{25.62 + \frac{49513}{7.639 + .72}} = .01048 \text{ lbs.;}$$

hence, $9130.61 \times .01048 = 95.688$ lbs. steam retained by cushioning. Net steam consumed per hour,

$$2566.043 - 95.688 = 2470.355 \text{ lbs.;}$$

and steam (water) per indicated horse-power per hour by the diagrams, $\frac{2470.355}{178} = 13.878$ lbs. The effective vacuum was [illegible] inches, and the losses by leakage and extra condensation were estimated as probably 15 per centum, hence

$$\frac{13.878}{.85} = 16.327 \text{ lbs.},$$

estimating an evaporative efficiency of connected boilers of 9 to 1 of coal; the cost of coal per I. H. P. per hour becomes [illegible] lbs. This is probably too low an estimate of this waste. Taking it, however, as even double this amount, 30 per cent., the coal and water consumption would be, respectively, but 2.2 and [illegible] lbs. per I. H. P. per hour; low figures, both.

The next illustration, a diagram published by Mr. Porter, as taken from a high-service pumping-engine at Providence, R. I., when making but one revolution per minute, exhibits the enormous extent to which initial condensation and later re-evaporation can occur, most remarkably.

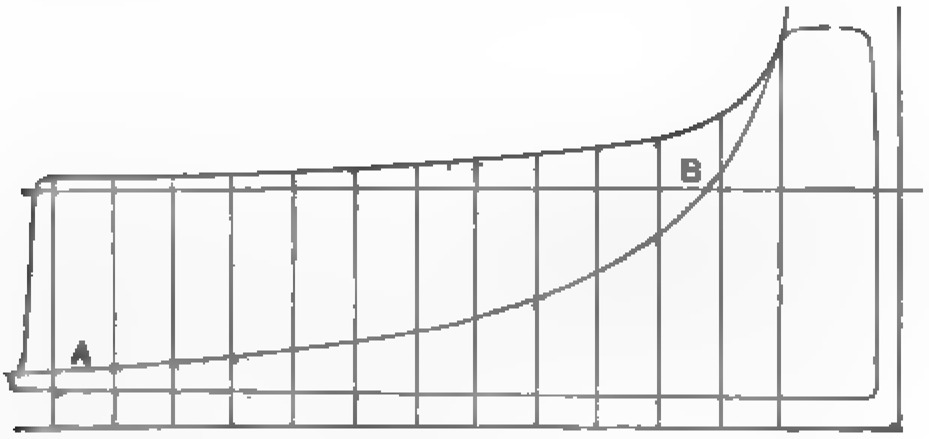

FIG. 105.—CONDENSATION AND RE-EVAPORATION.

The hyperbolic line is at *AB*, and the magnitude of the terminal ordinate of the diagram, as compared with the ordinate of the hyperbola, measures the proportion of re-evaporation. It is seen that more than three times as much steam must have been condensed at entrance as remained, to produce the diagram, this proportion, at least, being later re-evaporated.

The following are the quantities of steam found at various

parts of the stroke of a compound Corliss engine, as reported by Mr. Hoadley: *

	Steam, lbs.	
H. P. Cut-off.	Present.	Condensed.
0.178	9.97	6.80
.625	11.32	5.18
.750	11.35	5.15
1.000	11.35	5.15
L. P. Cyl., end.	10.57	5.93

The condensation in cylinder and jackets was about one-half throughout.

The next diagram, Fig. 106, illustrates the action of the air-compressor. The isothermal lines, which are here hyperbolic, are drawn from the atmospheric line as its starting-point. Two diagrams are shown superposed—the one of a common and somewhat inefficient compressor, the other of a more per-

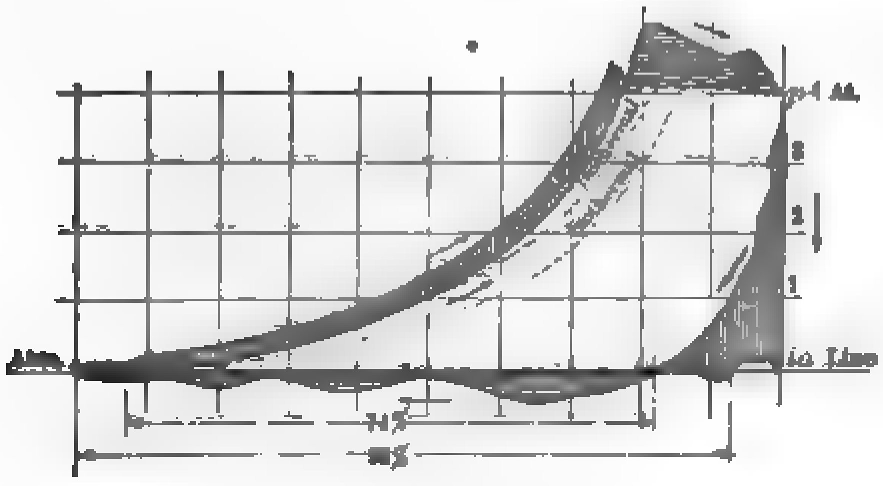

FIG. 106.—AIR-COMPRESSOR CARDS.

fect form. The former gives a diagram having an efficiency but 74 per cent. of the ideal, and the latter 93 per cent. Here the actual diagrams exceed the ideal in area, the heat of compression carrying its compression-line above the isothermal, and the defects of construction and operation of the induction and eduction valves throwing the delivery-line above the limit of pressures in the receiving reservoir.

* Steam-engine Practice in the U. S., 1884.

CHAPTER VII.

ENGINE-FRICTION, DYNAMOMETERS.

72. **Engine-friction** is an important element of waste in all engines. The resistance of the engine due to internal friction and the effort demanded for its impulsion are measured, according to size, type, and condition, by from about one pound on the square inch of piston in the best large and well-designed engines, to three or four pounds, and even more, in small and inefficient machines. An efficiency of machine exceeding 90 per cent. is rare, and is considered high.

The efficiency of mechanism, the ratio of work done by the engine to the work performed by the steam on its piston, is in rare cases 95 per cent., in usual good practice about 85 or 90, and in fairly good work 85 per cent., or less. If the figure falls under 0.80, it is regarded as decidedly low.

Before and during a trial, especially, the lubrication should be seen to be thoroughly efficient. A good lubricant should be chosen, and it should be properly applied and freely used.

73. **Indicated Work less Engine-friction** constitutes the net useful work of the machine. The power of the engine measured, not in its working cylinder, but as delivered from its crank-shaft, is that with which its proprietor and the engineer are most concerned. A complete engine-trial, therefore, includes a careful and exact measurement of the friction of the engine, and the so-called dynamometric power of the machine, as well as the indicated power recorded in the diagram. The friction of engine is sometimes allowed for when its direct measurement is impracticable, by assuming a certain pressure as suf-

ficient to overcome its resistance, which pressure ranges from one or two pounds per square inch in large engines, to three or four in smaller sizes, the experience and judgment of the observer being taken as guides. It has been found that engine-friction may usually be taken as constant at all loads.

In his various papers on this subject, the Author first called attention to the fact that the variation of load in steam-engines is not productive either of the method or of the amount of engine-friction that has been commonly assumed by earlier authorities on that subject.* It was shown that the formula of De Pambour, which makes the internal friction of the engine proportional to the load on its piston, is not usually correct, and probably is never so, with any familiar form of engine, or under any conditions often met with in practice. It was further shown that, under the conditions of usual practice, and at all ordinary speeds and pressures of steam, the resistance of the engine itself, its internal friction, remains sensibly constant, and that the so-called friction-card of the machine represents practically the friction of the engine when fully loaded, the indicated power without load being sensibly the measure of the wasted work of the engine when in operation under load of whatever amount. Even the compound engine, contrary to the expectation of the Author, exhibited substantially the same internal friction at all loads up to its full rated power, and with no load at all. He has shown the engine-friction to be independent of the load, but to be a function of the characteristics of the engine itself, of the speed of piston and rotation, of the steam-pressure, and of the method of steam-distribution, the two last-named conditions having slight effect, the others being most important. The weight and design, and the character of the workmanship of the engine, primarily determine the amount of its internal friction; the resistance is also a direct function of its speed, and it is slightly and observably affected, within

* Friction of Non-condensing Engines. Trans. Am. S. M. E., Vol. VIII, No. CCXXVIII, and Vol. IX, No. CCLXV.

limits, by the steam-pressure variations, and by the character of valve-gear and of steam distribution and of regulation of engine. The speed and weight of the running parts of the engine may, so far as can now be ascertained, be taken as the elements controlling friction of the machine. This leads to the conclusion that the friction coefficient of the rubbing surfaces decreases with the load on the engine and with increase of pressure on them, a result confirmed by numberless experiments of the Author and others, independently. With good lubrication, the coefficient of friction rapidly decreases with intensifying pressures, and to such an extent as to make the actual resistance to movement often very nearly constant.

74. **Measurements of Gross and Net Power** are commonly made by means of the indicator and either the absorbing or the transmitting dynamometer, the former giving the gross or indicated power, the latter showing the amount of power applied by the engine to a brake or to its special purposes, and capable of doing useful work. It is only when such measurements of actually applied, net, power are made, that the real value of the engine as a motor can be ascertained. The efficiency of the engine as a machine, also, is thus determinable, and is measured by the ratio of the dynamometric to the indicated power; this ratio is usually about 80 per cent., but sometimes exceeds 90.

Transmitting Dynamometers are of various types and forms, but all consist of a set of pulleys so arranged that they may be placed between the prime motor and the machinery to be driven by it, while the effort is measured by, usually, a set of springs interposed between the receiving and the delivering pulleys. The magnitude of this effort is often, perhaps generally, automatically recorded on a travelling ribbon or strip of paper, and the speed of the machine is observed. The product of the effort into the velocity of the point at which it is measured is the measure of the work done in the unit of time and of the power expended. There are many forms of this instrument, but the class most generally known is probably that of

General Morin, as built in the Sibley College shops, Fig. 107, in which *A* is a pulley fast on the shaft, and *C* is a loose pulley on the same shaft, the motion being transmitted from the prime mover to one or other, according as it is desired to drive the shaft or not.

A pulley, *B*, on the same shaft, carries the belt which transmits motion to the driven machine. This pulley is loose on the shaft, so that it is capable of moving backwards and forwards through a small arc, to admit of the deflection of a spring by which the effort is transmitted from the shaft to the pulley. One end of that spring is fixed so that the blade projects like

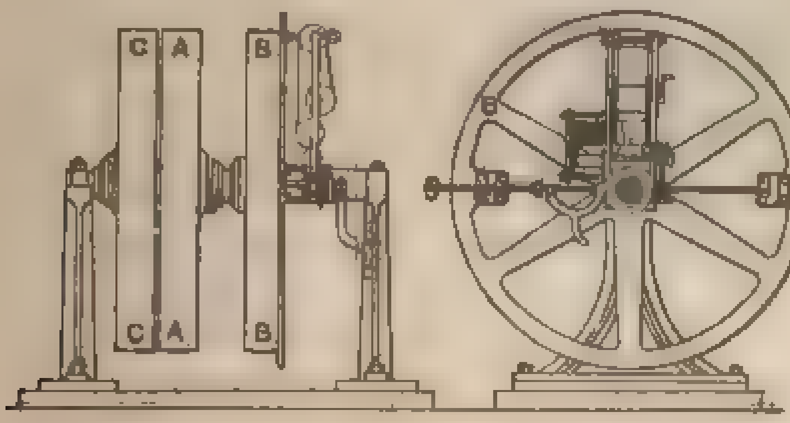

FIG. 107.—MORIN DYNAMOMETER.

an arm, and rotates with it. The other end is connected with *B*, so that the spring undergoes deflection proportional to the effort exerted by the shaft on the pulley. A frame, rotating along with it, carries an apparatus for making a band of paper move radially with a velocity proportional to the speed with which it rotates. A pencil carried by this frame traces a zero line, and another pencil carried by the spring traces a line whose ordinates represent the forces exerted. The mechanism for moving the paper is driven by a toothed ring surrounding the shaft, and kept at rest while the shaft rotates by means of a catch. When that catch is drawn back, the toothed ring rotates with the machine, and the paper is thus stopped when desired.

The Batchelder or Francis dynamometer, as designed by

Mr. Webber, is of the form shown in the following illustration. The principle of this machine was originally invented by a Mr. Samuel White, in England, in 1780–90; and was brought over to this country by Mr. Samuel Batchelder in 1836. It is said that one of these machines has been used fifteen years, weighing over one hundred and fifty thousand horse-power in small amounts, and the totals, in some cases aggregating two hundred and fifty horse-power, have substantially agreed with the results obtained from indicator-cards taken from the engine driving the same machinery.

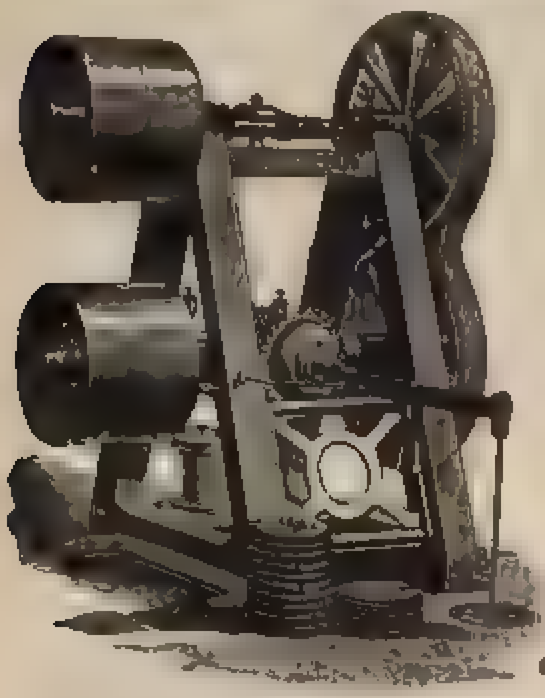
FIG. 108.—WEBBER'S DYNAMOMETER.

A transmitting dynamometer, another of many forms now obtainable, is shown in the next figure. This form is often employed for light work, as in determining in detail the power consumed by each of the several machines driven by the prime motor; and this has also been used by the Author with satisfactory results. As here seen, the pulley *A* is loose on the shaft, and receives the power. Its connection with the shaft is made by means of the spider *J*, which is keyed or screwed firmly to the shaft in close contiguity with the receiving pulley, its hub, in fact, forming one of the guides to the position of the pulley on the shaft. To connect this spider with the loose receiving pulley *A*, a bell-crank lever is pivoted into projecting ears on the rim of the wheel *A*, on opposite sides, the long arm of which connects with an annular slotted collar on the shaft by means of short bars *B*, *B*. The short arms of the bell-crank levers connect on the inside of the fixed wheel with two radial bars, one parallel to the outer arm of the bell-crank, and the other at right angles to it, receiving near its upper end a pivot passing through a swivel hung to the arm of the spider wheel, and having its extreme end pivoted to a stud fixed on the inner side of the

rim of the receiving pulley. It will be seen that the strain of the power received through the belt on the pulley will necessarily react on the levers, and through them on the spider, which may be considered nothing more nor less than a support to these levers in sustaining them in position to connect the loose receiving pulley with the shaft. It will be seen that the levers are connected by pivots with the sliding collar *BC*, in

FIG. 109.—THE EMERSON DYNAMOMETER.

the annular groove of which is seated a strap, with which is connected a forked lever, *CH*. To the end of the long arm of this lever a rod, *F*, with a short section of machine-chain is attached. This chain runs over the cylindrical head *D* of a pendulum weight, having a pointer, *E*, that traverses a fixed quadrant, *F*, properly divided by a scale to denote the relative pressure exerted through the medium of the receiving pulley on the shaft; the motions are absolute, there being no chance for "backlash."

75. **Calibration** of a transmitting dynamometer, as an example, one of the Morin type, is illustrated in the following:*

* Distribution of Internal Friction of Engines; R. H. Thurston; Trans. Am. Soc. M. E., Oct. 1888.

The best result in this work was given by such a dynamometer built in the Sibley College shops, at Cornell University. Its action is like that above described and is shown clearly by Fig. 110. A pulley, of which the rim, *B*, is shown, is fitted loose on the shaft, *S*. Four flat springs are securely bolted to the shaft, *S*, and to the rim, *B*. Now, if force be applied by a belt around *B* to turn the pulley, and if resistance

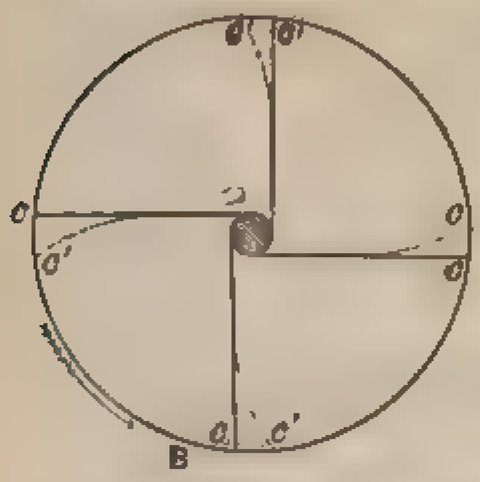

FIG. 110.

to its turning be produced by a fixed pulley on the shaft, *S*, from which some machine is driven by the belting, the springs, *c*, will be deflected into new positions, *c'*, an amount proportional to the force, and the fixed pulley will then revolve, thus driving the machine. To show the amount of power transmitted, and any variation that may occur in that power, a pencil is attached to the rim of the pulley, or to a post having an equivalent motion, and a recording apparatus, consisting of a series of gear wheels actuated by a spiral thread on a sleeve on the axis, causes a band of paper to move radially under the pencil. The recording apparatus can be stopped or started at will, without interfering with the motion of the machinery, by causing the loose sleeve to engage with a lug on the shaft. The diagrams obtained from the dynamometer consisted of a series of waving lines (Fig. 111) of varying elevation and with different average ordinates. The undulations were produced by changes of speeds probably caused by the inequalities of belt-lacings, etc.

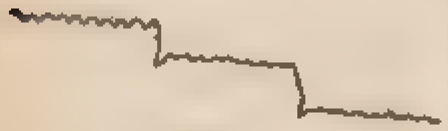

FIG. 111.

The dynamometer was calibrated in three ways: First, by attaching a brake to the same shaft, and comparing the diagrams with the brake-readings; secondly, by direct pull with a spring-balance against the springs of the dynamometer, and thus obtaining the ordinate for a given belt-pull; thirdly, on the same principle as the first, but a spring-balance was used, to measure the brake-weights, instead of scales. The object

of these calibrations was to obtain the ordinate corresponding to any given belt-pull. The following results were obtained, in the last case, by direct pull against the springs of the dynamometer, which method being employed, gave uniform and satisfactory results.

Pull on Dynam Pulley. Pounds.	Ordinate. Inches.	Pull on Dynam Pulley. Pounds.	Ordinate Inches.
0	0.40	35	1.80
5	0.65	40	2.08
10	0.80	45	2.32
15	1.02	50	2.58
25	1.33	60	3.08
30	1.55	70	3.52

The mean of three results corresponds very closely to the last, and, when plotted, gives a straight line, whose equation is $Y = 0.046X + 0.20$, Y being expressed in inches and X in pounds. According to General Morin, a good dynamometer should have (1) sensibility properly proportioned to the efforts to be measured; (2) the indications should be placed beyond the influence of the observer and given by the instrument; (3) the observer should be able to measure the effect at every point of the curve produced by the machine; (4) the apparatus should be constructed to give the total amount of work.

76. The Prony Brake, the Absorbing Dynamometer, or the Dynamometric Brake, has many forms.—A simple form of dynamometric brake for small powers is illustrated in Fig. 112. A is the shaft of the motor of which the power is to be determined; B is the pulley or drum on which the brake-blocks are changed by the bolts, C, C. The lugs, D, D, limit the movement of the beam, E, which is counterbalanced by a fixed poise at F, and the weights equilibrating the effort of the motor are applied at G.

Basswood and poplar are excellent woods for use in the rubbing parts of the dynamometric brake; but any wood will work well if properly handled. The soft are usually found better than the hard woods, but white ash and maple are good. End-grain is often preferred for rubbing surfaces. The wood

may be secured either to the pulley or to the strap of the brake. Where to be much or continuously used under heavy wear, it is perhaps better to put the blocks on the wheel.

The dynamometric power may be ascertained by means of a *rope-brake* upon the fly-wheel of the engine. Two ropes are

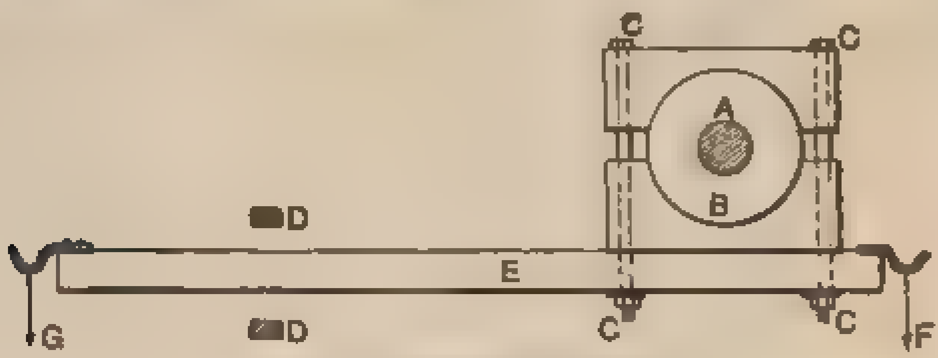

FIG. 112.—DYNAMOMETRIC BRAKE.

used for each wheel, kept at proper distances apart on the wheel by means of transverse wooden distance-pieces. The dead-load is usually applied by means of weights, and the back tensions necessary by means of a spring-balance. The spring-balance tension is deducted from the dead-load applied. This brake is found to work perfectly satisfactorily. If any metal be used for attaching the wooden cross-pieces to the ropes, it must not rub against the rim of the wheel; if this happens, the metal becomes hot, and is liable to burn the rope.

If it is assumed that L = length (effective) of the arm of the dynamometer in feet, W = weight (unbalanced) suspended on the arm in pounds, N = number of revolutions per minute, the horse-power will be

$$\text{D. H. P.} = \frac{2\pi WLN}{33000} = 0.0001904 WLN.$$

It is not unusual to make the effective value of r equal to $\frac{33}{2\pi}$; so that, the circumference described being 33 feet, the power is at once completed by multiplying the weight and number of revolutions of the shaft, W and N, together, and dividing by 1000 to get the horse-power.

A form of this brake which the Author has most frequently employed, and with satisfaction, is constructed as follows: *

Like nearly all dynamometers of this class, it includes a brake-wheel, or pulley, which is keyed on the engine-shaft, and is sufficiently strong to sustain safely the maximum load anticipated. The rim of this pulley is turned flat and smooth, and fitted with a flexible brake-strap of wrought-iron, or other suitable material, which may be adjusted to such a tension as will enable it to control the engine at maximum power. In this case, the rim is trough-shaped in section, flanges extending inward toward the shaft to a sufficient depth to permit the retention in the circular trough so formed of a stream of water which is used to keep the pulley cool, and to carry away the heat produced by transformation of mechanical energy. The two ends of the brake-strap are united by a right- and left-hand screw, in such manner that they may be drawn together and the strap set up to any desired degree of tension. The brake-arms consist of two beams of wood, forming a < frame, and secured to the strap at the upper and lower sides, and at their junction supported by a strut resting on a platform-scale of nice construction and great accuracy. As the engine-shaft revolves, the tendency of the brake-arms to turn is resisted by the scale; and the effort so measured, multiplied by the relative velocity of the engine-shaft and the supported point on the arm, gives a measure of the power expended. Water is supplied to the pulley-rim, by means of a hose, from any convenient source, and the excess is taken away in a similar manner. The centrifugal action of the rotating mass keeps the fluid in place in the pulley-rim, and the eduction-pipe receives the water carried away by it as the tender of a locomotive scoops water from between the tracks, when at high speed. This system permits efficient lubrication, without admixture of grease with the water, and secures a perfection of smoothness and uniformity of rubbing-surfaces unattainable with older forms of brake.

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* Construction of a Prony Brake; R. H. Thurston; Journal Franklin Institute, April 1886.

77. Designing a Brake.—The following is an account of the design of a brake, which worked well under higher engine loads than the Author had ever before known to be controlled by this means.* It also illustrates fully its theory.

The brake was designed for the maximum power of the engine, *i.e.* taking steam at full stroke, the engine running at 100 pounds pressure, and at 100 revolutions per minute. The diameter of the cylinder was 18 inches and the stroke 42 inches, and we have for the maximum power developed

$$\text{H. P.} = \frac{254.47 \times 100 \times 42 \times 2 \times 100}{33000} = 540 +$$

The brake was accordingly designed to control the engine when exerting this power, and to be used upon a pulley of 5 feet diameter and 24-inch face. The size of the pulley was chosen of this diameter, simply because it compelled less removal of floor and railings about the engine, and would also cost less than a larger one.

The calculations for the remaining parts of the controlling apparatus is as follows:

Assumed diameter, 5 feet; assumed maximum speed of engine, 100 revolutions; circumference, 15.708 feet. This would give for the greatest linear velocity of the pulley per minute, 1570.8 feet. Dividing the number of foot-pounds developed by the engine at its maximum speed and pressure, by the linear velocity, gives the resistance at the rim of the pulley

$$\frac{540 \times 33000}{1570.8} = 11345 \text{ pounds};$$

which figure is the total friction, in pounds, on the face of the pulley.

The brake-blocks were $2\frac{1}{2}$ inches thick, 5 inches wide, and 24 inches long, of unseasoned white oak, and were placed 7 inches from centre to centre, leaving a space of 2 inches, between adjacent blocks, for diffusion of the heat and for lubrication. The blocks were attached to the flexible brake-straps

* The designers were Messrs. Gately and Kletsch.

by means of wrought-iron lag-screws. The three blocks at the top and those at the bottom of the pulley were fastened to the arms of the brake.

The straps, two in number, were calculated thus:

Let T_1 and T_2 represent the tensions at the ends of the band which embraces the pulley, and let T_1 be the maximum tension.

Then T_1 exceeds the tension T_2 by an amount equal to the friction between the blocks and the pulley; *i.e.*,

$$R = T_1 - T_2 = 11345.$$

Let c denote the ratio which the arc of contact bears to the circumference of the pulley, f the coefficient of friction between the blocks and the pulley; then the ratio $T_1 : T_2$ is the number whose common logarithm is $2.7288cf$; or,

$$\frac{T_1}{T_2} = 10^{2.7288fc} = N.$$ (Cotterill p. 236)

c, the arc of contact of the bands, $= 1$, and f, the coefficient of friction between wood and cast-iron (well lubricated), was taken at 0.2; then

$$N = 10^{2.7288fc} = 10^{2.7288 \times .2 \times 1};$$

or,

$$\frac{T_1}{T_2} = 10^{0.54576} = 3.5.$$

Having found $R = 11345$ pounds, we have for the greatest tension on the band

$$T_1 = R\frac{N}{N - 1};$$

and substituting the values of R and N in this equation, we have

$$T_1 = 11345\frac{3.5}{3.5 - 1} = 15883 \text{ pounds}.$$

Hence, for the combined tension on the band, and using two straps, we have for the tension on one

$$\frac{15883}{2} = 7941.5 \text{ pounds.}$$

Taking the tensile strength of such wrought-iron as safe at 40,000 pounds per square inch, and allowing for a sixfold factor of safety, we obtain for the section of the band

$$\frac{7941.5 \times 6}{40000} = 1.19 \text{ square inches.}$$

The nearest band-iron of this section was $\frac{3}{8} \times 3$ inches, and, after careful testing, it was found to be of sufficient strength; giving, at the same time, that flexibility which is of vital importance in the operation of brakes. At each end of the bands it was found necessary to weld on round bar-iron of equal section, to admit of threads being cut for the purpose of tightening and loosening the brake.

The arms were two in number, of 6×6 inches well-seasoned spruce. The length was made 10 feet 6.1 inches from centre of the bearing-surface on the pulley to centre of bearing-surface on the scale, as it brought the scale beyond the rim of the fly-wheel, and also greatly facilitated calculations of the horse-power developed—the circumference of a circle, whose radius is 10 feet 6.1 inches, being 66 feet. Thus instead of multiplying by 66 feet, and then dividing by 33,000 to obtain the horse-power, it is only necessary to divide the product of the net scale-pressure and the revolutions per minute by 550, the quotient being the horse-power developed, *i.e.*,

$$H.\,P. = \frac{W \times Rev. \times 66}{33000} = \frac{W \times Rev.}{500}.$$

The stand through which the pressure was transmitted to the scale was composed of two uprights, 6×6 inches, of white pine, surmounting a pedestal covering the greater part of the

scale platform. Upon these uprights was placed a steel plate of $\frac{3}{4}$-inch thickness, which received the pressure of the bolts. The scale was carefully balanced, and was capable of accurately weighing 3000 pounds. All weights used were carefully weighed on a standard balance, and none were used that were found not to be absolutely correct.

As the common segmental arm would give but a very narrow bearing for the rim, the Author advised an arm of I-section, which was found to answer the purpose.

The calculations for the parts of the pulley were made according to Unwin,* giving for the thickness of rim

$$t = 0.7\delta + 0.005D = 0.65 \text{ inch};$$

where $D =$ diameter in inches $= 60$ inches;
and $\delta =$ thickness of belt taken at 0.5 inches.

The number of arms was assumed at 6; and similarly, for the thickness at the nave,

$$h = 0.1781 \sqrt[3]{\frac{PD}{n}} = 8.54 \text{ inches};$$

P being the driving effort, 11345 pounds;
$D =$ diameter $= 60$ inches; and
$n =$ number of arms $= 6$;
$h_1 =$ breadth of arms $= \frac{h}{2} = 4.27$ inches.

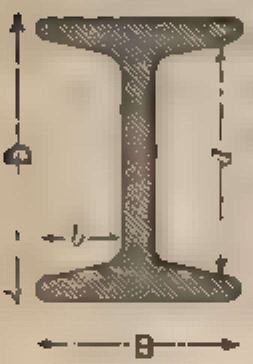
For h at the rim, we take $\frac{2}{3}$ the diameter of the nave.
For the thickness of the nave,

$$\delta = 0.18 \sqrt[3]{BD} + \tfrac{1}{4} = 2.1 \text{ inches};$$

where B is the face $= 24$ inches;
D is the diameter $= 60$ inches.

* Machine Design.

The diameter of the main shaft being 9.12 inches, the calculated thickness of the nave was judged rather small, and 2.5 was used instead. The rim was also made $\frac{1}{8}$ inch heavier at the centre than the calculated dimension used for the edges of the rim.



For the moment of inertia of this section of arm,

$$I = \tfrac{1}{12}BD^3 - \tfrac{1}{12}(2bd^3);$$

and, considering the arm as fixed at one end and loaded at the other,

$$Pl = \frac{RI}{d}; \quad R = \frac{Pld}{I};$$

where $P =$ load;
$l =$ length of arm;
$I =$ moment of inertia;
$R =$ modulus of rupture;
$d = \frac{1}{2}D$.

Load on 1 arm $= \frac{1}{6}$ of $15600 = 2600 = P$.
$l = 30 - 7\frac{1}{2} = 22.5$ inches.
$I = \frac{1}{12}(4 \times 8^3) - \frac{1}{12}(2 \times 1\frac{1}{2} \times 5^3) = 40.6$

Allowing a factor of safety of 8, we have

$$R = 17290,$$

and the above sections and dimensions are ample.

The standardization of this type of absorbing dynamometer consists simply of the careful determination of its exact dimensions, and of the accuracy of the scales employed.

In the original device of Prony, the efforts on the brake were obtained by loading a scale-pan, a far less manageable system than that which is above described. Many modifications of this instrument have been made by various ingenious engineers, some of which may be here briefly referred to.* In all cases and in all applications of the brake, the essential conditions are, mainly, accuracy of measurement and readings, and,

* See Beaumont on Friction Brake Dynamometers, Proceed. Brit. Inst. C. E., 1888, for a very full account of these instruments.

above all else, uniform friction-resistance, such as can only be insured by good methods of lubrication. All conditions of the trial should be as constant as possible. For small powers, the arm may be omitted and the restraining effort exerted directly upon the strap, and this strap is sometimes merely a leather band or a rope, either with or without blocks. In all cases the work in actual horse-power with the load P in pounds at the distance R in feet from the centre of the wheel, making N revolutions in time t in minutes, will be

$$H.P. = \frac{(R \times 2) \times \pi \times P \times N}{33000 \times t};$$

or taking c = circumference of circle of radius R, and V = velocity in feet per minute of circumference, then $V = cN$, and

$$H.P. = \frac{cNP}{33000t} = \frac{VP}{33000t},$$

$$P = \frac{33000\ H.P.}{cN} = \frac{33000\ H.P.}{V},$$

$$c = \frac{33000\ H.P.}{NP}, \qquad N = \frac{33000\ H.P.}{cP}.$$

In many forms of brake, it is sought to secure a self-adjustment of the grasp of the strap, as, for example, the dynamometer of Messrs. Amos and Appold, Fig. 113. It is provided with a compensating lever, K, by means of which the rise or fall of the load P is attended with a decrease or increase in tension on the brake-strap. With a given tension in the belt, and with the load P set with its point of suspension H opposite the point T, the lever takes a vertical position; but as soon as the load P is lifted the lever pivoted at X moves with and virtually increases the length of the belt, and thus slackens it, allowing the load again to descend. If the total friction decreases, the descent of the load carries the compensating lever

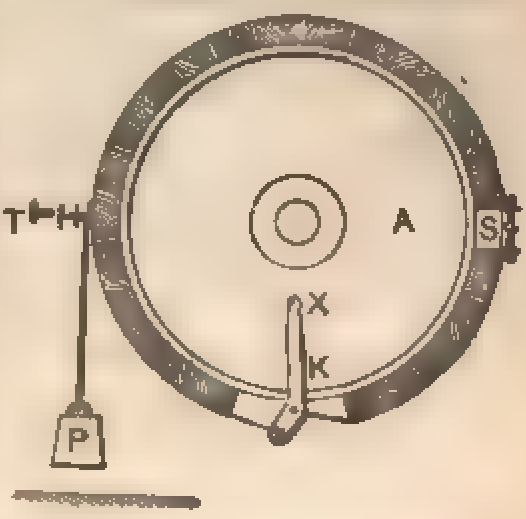


FIG. 113.—COMPENSATING BRAKE

to a new position, thus tightening the belt and increasing the friction.

In a form of compensating brake designed by Mr. Balk, Fig. 114, the compensating lever is outside the circumference of the strap. It is connected at B and at C to the strap, and a fixed pin, F, passes through a slot at the outer end, where is suspended a weight, p, sufficient to keep the lever free. This

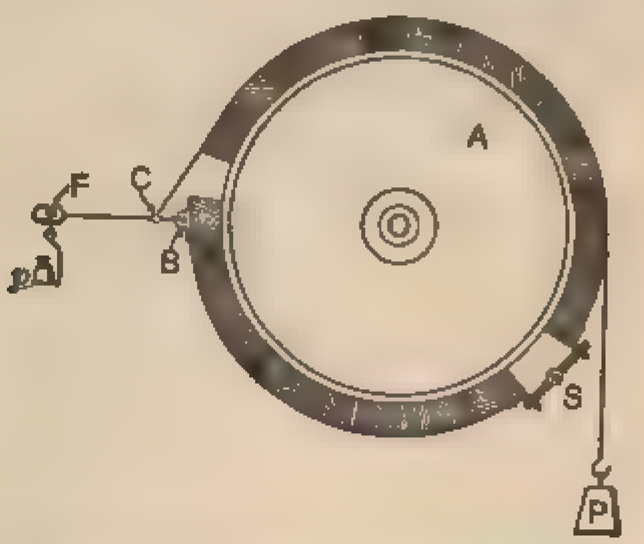


FIG. 114.—COMPENSATED DYNAMOMETER.

weight must be varied with change of condition of the brake-blocks, the lubricant and the temperature of the wheel, and as it must be taken as acting at the radius OF in favor of the weight P, these variations become troublesome by virtually making P a variant.*

The form of pulley which has been described is illustrated in the next figure, which shows the proportions adopted by Mr. Halpin, who was probably the first to use it.

Mr. Beaumont has proposed the following expression as a convenient means of comparing the relative capacity of brakes, judging by the amount of work for which they have been designed, or to which they have been put. The coefficient $K = \frac{WV'}{HP}$ may be employed, W being the width of the wheel

* See paper by Professor Brauer, Zeits. der Ver. Deutschen Ing., Band xxxii; Seite 56.

in inches, and V' = the velocity of the periphery of the wheel in feet per minute. This gives for the

Royal Agricultural Society's single brake	$K = 824$
" " " treble "	$K = 495$
Garrett's water-cooled brake	$K = 740$
Ransomes, Balk's brake	$K = 1020$

Compared in this same manner the Prony brake described by the Author, as devised for measuring a maximum of 540 H. P., the wheel being 5 feet in diameter and 2 feet wide, and $N = 100$, gives $K =$ only 75. This brake was freely lubricated with beef

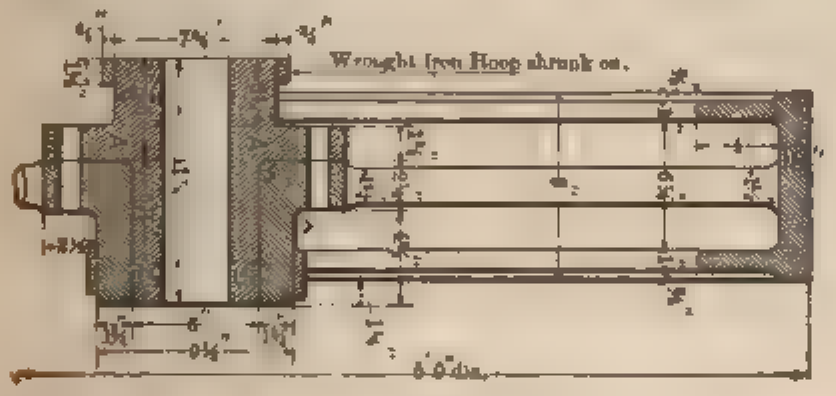


FIG. 115.—BRAKE PULLEY.

tallow, plumbago, and lard oil, and although designed for a maximum of 540 H. P., it was not worked above 200 H. P., and at this power $K = 188$. The more effective the brake the lower the coefficient. $K = 400$ may be taken as a good figure; the coefficient is less, apparently, on the water-cooled hollow rim. Other things equal, the highest coefficients correspond to largest areas of rubbing-surface on the rim. Mr. Beaumont makes the following approximate estimate of the maximum pressure p per square foot of block. W' being the width of the wheel in feet, p may be obtained thus:

$$T + T = DpW', \quad \text{or} \quad T = \frac{DpW'}{2},$$

and

$$p = \frac{T + T}{DW'}.$$

The greatest pressure per square foot of surface of the Author's example of a Prony brake, if it had been used as proposed for 540 H. P., would have been, at $f = 0.2$,

$$p = \frac{15883 \times 2}{5 \times 2} = 3176 \text{ lbs.}$$

At 200 H. P. $p = 1180$ lbs. The greatest pressure per square foot on the blocks of the Royal Agricultural Society's brake at Newcastle was $p = 312$ lbs. with the blocks covering 0.8 of the surface of the wheel.

This is a rough approximation to the greatest pressure per square foot, but it affords a fair means of comparison of the pressure with the different brakes.

A good method of fitting the dynamometric brake to a portable engine is shown in the illustration.* *A* is the beam of the brake; *B* is the point of attachment of a strut which transmits the pressure due the turning effort to the platform-scale, seen below; *C* is a counterbalance.

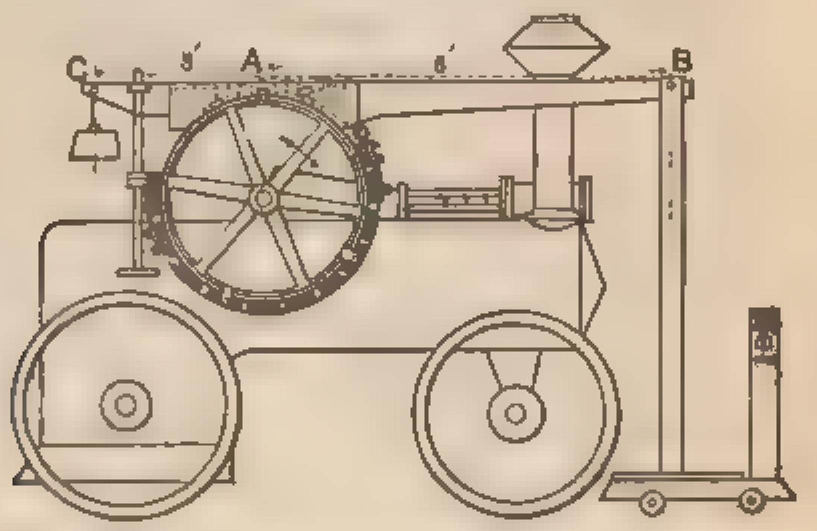


FIG. 116.—DYNAMOMETRIC BRAKE.

In handling a brake cooled by water, it is advisable to use as small a quantity of the liquid as possible, and it will often be found perfectly practicable to even permit its complete evaporation, keeping the metal at 212° F., thus reducing the

* On the Distribution of Internal Friction of Engines; R. H. Thurston; Trans. Am. Soc. M. E., 1888.

quantity demanded to a small fraction of that often used, and largely doing away with the difficulties incident to its supply and wholly with those of discharge.

In the Brauer dynamometric brake,* Fig. 117, instead of wooden jaws hugging the rims of the pulley, an iron band is used for flat-rim pulleys, and wire-ropes for grooved pulleys. The apparatus is composed of the following parts:

1st. The iron band or wire-rope.

2d. The clutch-producing arrangement and the regulation of the tension of the clutch-producing bands or ropes.

The iron band applied to an ordinary flat-rim pulley is provided with four double guides k (Fig. 117) to retain the band in position, fixed up by means of a stirrup bolt and link and a safety cord $l'l'$, so as to allow the band to have a play of 100 millimetres. These safety measures can be modified according to circumstances.

The frictional parts are actuated in the following manner:

The wire band is attached at its ends to the point of the application of resistance a, and to the point of rotation b by a lever abc. The point c of this lever is attached at d by means of a winding tackle ee_1, and a spring f, to the upper end of the wire band. The cord of the winding tackle after leaving e is passed round the friction-roller g. The operator, by pulling the cord, produces the tightening action in such a way that the weight p is lifted and equilibrium established. In order that this condition may not be affected by the tension of the part of the cord from e_1 to g, it is only necessary to fix the friction-roller in such a way as to be in line with the axis of the pulley or drum. The moment of this tension will then be *nil*. The friction-roller is not indispensable, but its application permits the operator to control the action of the brake at some distance. Automatic regulation is effected, according to the variations of the friction, by the combined action of the spring f and the cord h. It will be seen that if, owing to excessive friction, the weight p is lifted above its mean position, the cord

* "Bulletin de la Société Industrielle de Mulhouse," 1884, p. 485. Proc. Inst. C. E., No. 2079.

h will be stretched, augmenting the tension, and elongating the spring *f*. The result is a diminution of the clutching action, and the weight *p* will be lowered.

The weight *l*, which obviates the necessity of attaching the cord *h* to the floor-boards, should be equal the strain liable to be brought upon the cord *h*.

The lifting-tackle should always be suspended in such a way as to reduce the traction upon the cord *h* to a minimum, and so that the influence of this factor upon the condition of equilibrium in general need not be taken into consideration.

Lubrication may be effected in any convenient manner. The rim is usually sufficiently cool under light load without water.

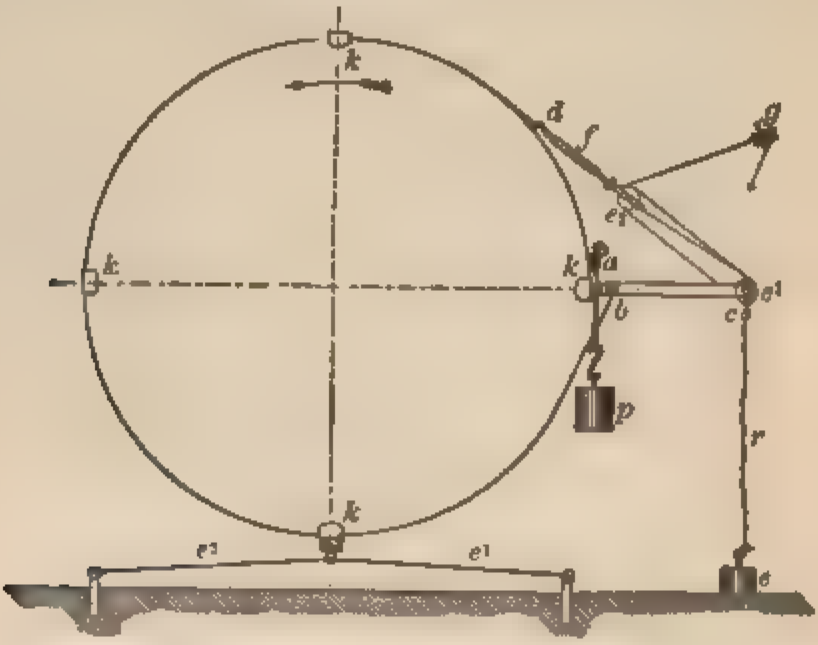


FIG. 117.—BRAUER'S BRAKE.

78. Data obtained by the use of the transmitting dynamometer are often of great value as a check on other methods of test. Mr. Emerson gives the following as illustrating this fact:

Testing woollen-mill machinery, running ten "sets," eleven hours per day, with the boiler-pressure kept at 70 pounds, the driving-pulley on engine, 9 feet diameter, with 30-inch double belt, drove a 5-feet pulley upon the main line. Throwing on and off machinery caused a variation of four revolutions of the pulley on the engine, or from 120 down to 116 per minute, 3½ per cent.

The results obtained were as follows:

Average net effort for 11 hours, ... 1163 lbs.
Coal burned in 11 hours, ... 4955 lbs.
Average power in 11 hours, ... 82.9 H. P.
$4955 \div 11 = 450.4 \div 82.9 = 5.43$ lbs. coal per horse-power per hour.

The St. Joseph Milling Co., Mishawaka, Ind., 100-barrel mill, required 3.81 horse-power per bushel.

The Ripple Mill, Mishawaka, Ind., 130-barrel mill, required 3.91 horse-power of water per bushel ground.

Mishawaka Mill, Mishawaka, Ind., 175-barrel mill, required 4.72 horse-power of water per bushel.

Sage Brothers' Flouring Mill, Elkhart, Ind., roller-mill, 280-barrel capacity, required 3.18 horse-power of water per bushel ground.

TESTS ON SPINNING FRAMES, SPEEDERS, ETC.

No. of Test	Time.	No. of Spindles.	No. of Yarn	No. of Weighings	Condition.	No. of Turns per Minute of Spindles. Heaviest.	Lightest.	Variation.	Average.	No. of Test.
1	2.00 till 5.30	192	28^2	6	Bobbins from ½ full till doffed	7343.5	7556.0	212.5	7449.2	1
"	"	"	"	3	Bobbins from doff till ½ full	7392.0	7343.5	9.3	7419.3	"
"	"	"	"	9	Total of above	7343.5	7343.5	0	7434.4	"
2	8.00 till 10.00	192	28^2	7	Bobbins full till ½ full again.	7228.5	7362.8	134.3	7287.2	2
3	10.30 till 11.30	192	28^2	6	Bobbins ½ full till full	7355.4	7246.8	108.6	7324.2	3

No. of Test	Horses-power to a Frame. Heaviest	Lightest	Variation	Average	Spindles per Horse-power. Heaviest	Lightest	Variation	Average.	No. of Test
1	1.7000	1.6150	0850	1.6660	113.0	119.0	6.0	115.3	1
"	1.5050	1.3820	1230	1.4420	127.7	138.0	9.3	132.4	"
"	1.7000	1.3820	3180	1.5915	113.0	138.0	25.0	120.7	"
2	1.3504	1.2500	1004	1.3106	142.2	153.6	11.4	146.5	2
3	1.6180	1.4205	1975	1.5094	118.6	135.0	16.4	127.2	3

Mr. S. Webber finds the friction of this class of dynamometers to be sensibly constant.

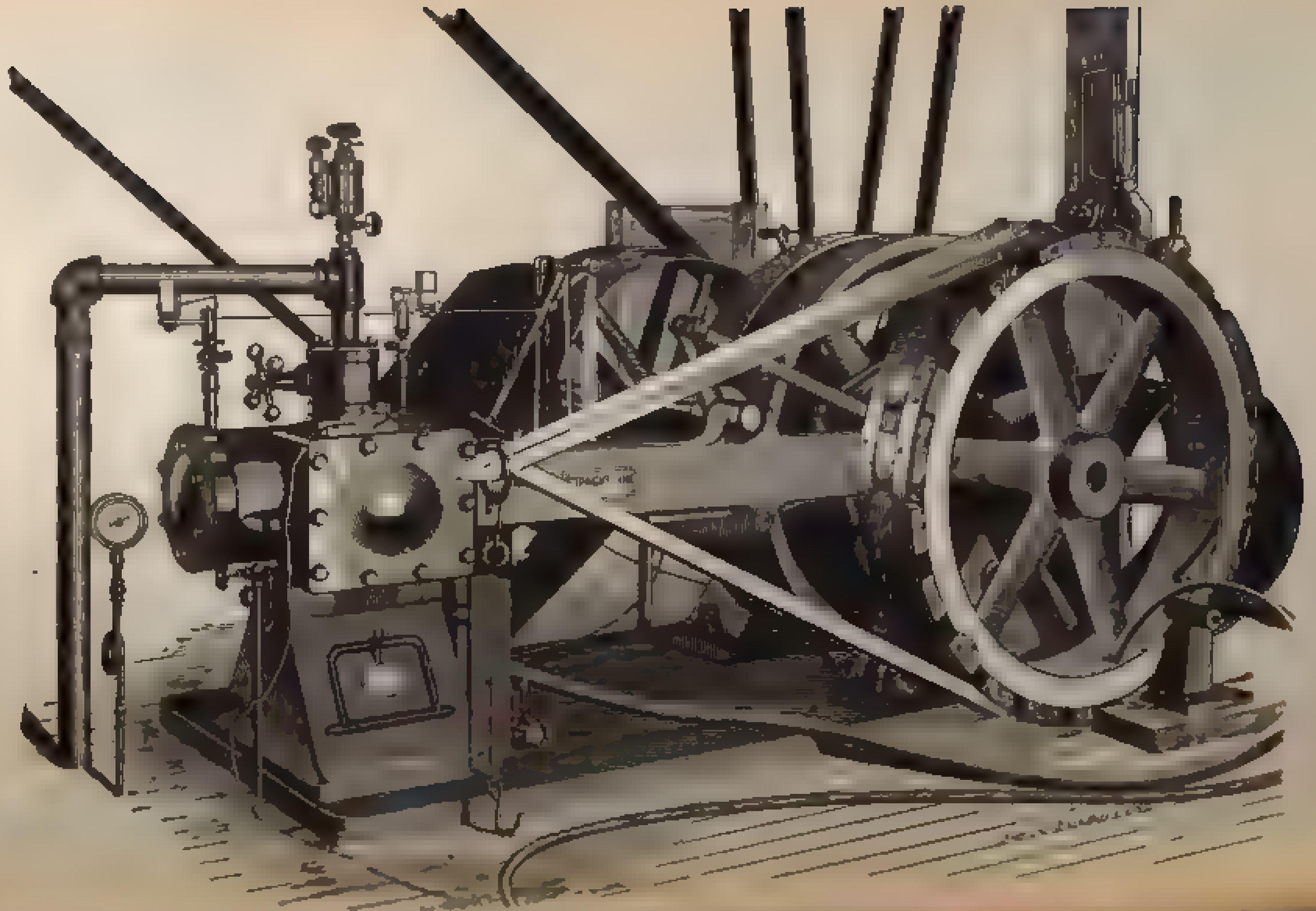


Fig. [illegible].—Fitting up the [illegible]

CHAPTER VIII.

STANDARD METHODS OF ENGINE TRIAL.

79. Standard Methods of Engine Trial have been proposed by various writers and practitioners, with the double purpose of securing all needed data, at least cost in time and money, and of making all results strictly comparable. In the absence of agreement in regard to method, a great variety of practice is liable to spring up, and as great a variety of methods of securing, tabulating, and computing results from the data obtained. It is therefore considered advisable that all such experimental work, whether for directly practical purposes or with a scientific object, should be made by that carefully planned and precisely stated system, which should be best adapted to the ready determination of all needed data, with least liability to error, and a most convenient means of checking all figures. Both engine and boiler trials should be made in accordance with such a system as is generally recognized as well adapted to its purposes, and accepted and indorsed by those whose learning and professional standing and experience best fit them to judge it. Standard engine and boiler trials are not yet as widely adopted as is desirable; but they are gradually taking definite form and are steadily coming into general use.

The results of any engine trial, if complete and accurate, should enable the engineer to answer several questions:

(1) What is the real efficiency and the economical performance of the system tested?

(2) How does it compare with standard apparatus of a similar character? and in what is it superior or inferior? What are its excellencies and its defects?

(3) How are commercial and financial conditions affected by its operation?

These questions being solved, the proprietor will know to what extent his expenditures and his methods of operation are wise and productive; the builder will learn how successful he has been in his work, where it is defective, and what remedy is available; the engineer secures data which enable him to design intelligently later and better constructions, and it may furnish a standard for still other comparisons.

The duty-measurement should always be expressed in perfectly definite terms. The usual expression may be interpreted to assume any one of several different units. If the efficiency of the system, engines and boilers included, is to be measured only, it is sufficient to ascertain the relation between the work performed and the cost of its performance as measured at the boilers; but even this may be a very uncertain measure unless the quality of the fuel is prescribed. The kind of fuel should be stated in all such cases; but a system of measurement which determines the heat produced in the furnace, in thermal units, or other equally definite terms, and the quantity of *useful* work which it yields, is the only satisfactory one.

The only correct and exact method of gauging the performance of any steam-engine is to determine the weight of steam or, better, the number of thermal units demanded by it per horse-power per hour. The proper measure of the boiler efficiency is the proportion of the heat of combustion of the fuel which is absorbed and stored as available energy in the steam which it produces. To rate the engine by the quantity of fuel burned at its boiler is wholly incorrect; to rate the boiler by the ratio of steam made to coal burned is hardly less indefinite. It is only by the habitual use of a known fuel of uniform composition and physical character that comparisons of value may be effected at all. Even where the steam-unit is adopted, it must be taken at a standard temperature and pressure. In all heat-engines the proper measure of heat-energy is the heat-unit. There is, therefore, reason in the adoption of, for an exam-

ple, 1000000 B. T. U., a figure sometimes so taken, as a standard quantity in duty-trials of engines.*

80. Engine and Boiler Trials are so generally conducted together, and the former so commonly depend for their essential data upon the latter, that the presentation of illustrations of standard methods for both is considered advisable and will be given later. Of the two, the latter is usually much the more laborious and troublesome, and also more expensive in time and money. The boiler-trial also admits of a greater variety of methods than the engine-trial, and is correspondingly more liable to yield results of varying accuracy. For this case the best practice is better settled than for the other, and standard methods of steam-boiler trial are fairly well established on both sides the Atlantic.

A good boiler should have an efficiency of not less than 75 per cent., giving thus 10875 B. T. U. per pound of carbon supplied as fuel, and, deducting ash, for good coal, about 10000 B. T. U. per pound of fuel, the equivalent of 0.25 pound of coal per horse-power and per hour, if all converted into work. The real efficiency of any engine is measured by the quotient of this quantity for the actual consumption. Thus, an engine using 2.5 pounds of good coal has an efficiency of heat

$$E = \frac{0.25}{2.50} = 10 \text{ per cent.}$$

This is considered a good result. Ninety per cent. of all heat supplied is here wasted. An engine using 1.25 pound of best fuel, a result sometimes claimed, but certainly seldom reached, has an efficiency of

$$E = \frac{0.25}{1.25} = 20 \text{ per cent.}$$

A consumption of 40 pounds of steam would give

$$E = \frac{2.5}{40} = 6 \text{ per cent.}$$

* This corresponds to 100 lbs. coal evaporating 11.25 lbs. water from and at 212° F.

As an example of a systematic scheme of engine-trial, illustrating the general characteristics of a standard method, we may take the following outline abstract from a plan for a pumping-engine duty-trial proposed by Mr. Barrus: *

(1) It is presumed at the outset that the engine is in thorough order in every part, having been in operation a sufficient length of time, since its erection, to secure easy and proper working. If this has not been done, and especially if the plunger has been recently packed or its packing newly re-adjusted, the engine is worked for a run of at least twelve hours' continuous service in preparation for the test.

(2) The plant is subjected to a preliminary run, under the conditions determined upon for the test, for a period of at least three hours, so as to find the temperature of the feed-water (or the several temperatures, if there is more than one supply), for use in the calculation of the duty. During this run the observations are made every fifteen minutes, and the results averaged.

(3) The engine is now stopped for a time, in order, first, to connect up the measuring apparatus for determining the weight of the feed-water consumed, or of the various supplies of water if there are more than one; and, second, to test the leakage of the plungers.

The quantity of water which leaks by the plungers is most satisfactorily determined by removing the cylinder-heads. A wide board or plank is temporarily bolted to the lower part of the end of the cylinder, so as to hold back the water, in the manner of a dam, and an opening is made in the temporary head, thus provided, for the reception of an overflow pipe. The plunger is blocked at some intermediate point in the stroke (or if this position is not practicable, at the end of the stroke), and the water from the force-main is admitted at full pressure behind it. The leakage escapes through the overflow pipe, and it is collected in barrels and measured. The test need not continue over fifteen minutes, or, if carefully made, a less time, the desired object being to get a satisfactory deter-

* London Engineering, Mch. 1, 1889, p. 217.

mination of simply the rate of leakage. If no means exists for putting the back side of the plunger under water-pressure, a suitable pipe can readily be provided for the purpose. Should the escape of water in the engine-room be objectionable, a spout may be constructed to carry it out of the building. Where the leakage is too great to be readily measured in barrels, or where other objections arise, resort may be had to weir or orifice measurement, the weir or orifice taking the place of the overflow pipe in the temporary wooden head. The apparatus may be constructed in a somewhat rude manner, and be sufficiently accurate for practical requirements.

In the case of a pump from which it is difficult to remove the cylinder-head, it may be desirable to take the leakage from one of the openings which are provided for the inspection of the suction-valves, the head being allowed to remain in place. If the test is made without removing the head, leakage of the discharge-valves may be confounded with leakage of the plunger. Examination for such leakage should be made first of all, and if it occurs and it is found to be due to disordered valves, it should be remedied before making the plunger-test. The discharge-valves on the back end of the pump should likewise be examined, as also the suction valves on both ends, and the disordered valves removed. Leakage of the discharge-valves will be shown by water passing down into the empty cylinder at either end when they are under pressure. Leakage of the suction-valves will be shown by the disappearance of water which covers them.

The leakage-test being completed, no change is allowed in the adjustment of the packing of the plunger (supposing this to be of a form capable of adjustment), the head is immediately replaced and preparations made for at once beginning the main duty-trial.

(4) The duty-trial is here assumed to apply to a complete plant, embracing a test of the performance of the boiler, as well as that of the engine. The test of the two will go on simultaneously after both are started, although the boiler-test will begin a short time previous to the commencement of the en-

gine-test, and continue after the engine-test is finished. The mode of procedure is as follows:

While the preparations are being made to start the engine, after the completion of the leakage-trial, steam is raised in the boiler to the working pressure. The fire is then hauled, the furnace and ash-pit cleaned, and the test of the boiler is commenced. This test is made in accordance with the rules for a standard method recommended by the Committee on Boiler-Tests of the American Society of Mechanical Engineers.* This method, briefly described, consists in starting the test with a new fire lighted with wood, the boiler being previously heated to its normal working degree; operating the boiler in accordance with the conditions determined upon, weighing coal, ashes, and feed-water; observing the draught, temperatures of feed-water and escaping gases, and such other data as may be incidentally desired; determining the quantity of moisture in the coal and in the steam: and at the close of the test hauling the fire and deducting from the weight of coal fired whatever unburned coal is contained in the refuse withdrawn from the furnace, the quantity of water in the boiler and the steam-pressure being the same as at the time of lighting the fire at the beginning of the test. The temperature of the feed-water is observed at the point where the water leaves the engine-heater if this be used, or at the point where it enters the flue-heater if this apparatus is employed. In either case, where an injector is used for supplying the water, a deduction is to be made for the increased temperature of the water due to this method of feeding.

As soon after the beginning of the boiler-test as practicable, the engine is started and preparations are made for the beginning of the engine-test. The formal commencement of this test is delayed till the plant is in normal working condition, which should be not over one hour after the time of lighting the fire. When the time for commencement arrives, the feed-water is momentarily shut off, and the water in the lower tank

* See this report as given in Chapter II of the present work.

is brought to a mark. Observations are then made of the number of tanks of water thus far supplied, the height of water in the gauge-glass, and the indication of the counter on the engine, after which the supply of feed-water is started and the regular observations of the test commenced. The test is to continue at least ten hours. At its expiration the feed-pump is again momentarily stopped, care having been taken to have the water slightly higher than at the start, and the water in the lower tank is brought to the mark. When the water in the gauge-glass has settled to the point which it occupied at the beginning, the time of day and the indication of the counter observed, together with the number of tanks of water thus far supplied, the engine-test is held to be finished. The engine continues to run after this time till the fire reaches a condition for hauling and completing the boiler-test. It is then stopped, and the final observations relating to the boiler test are taken

The observations to be made and data obtained for the purposes of the engine test embrace the weight of feed-water supplied by the main feeding apparatus, that of the water drained from the jackets, and any other water which is ordinarily supplied to the boiler, determined in the manner already pointed out. They also embrace the number of hours' duration and number of strokes of the pump during the test, as noted, together with the length of the stroke (in direct-acting engines), the indication of the gauge attached to the force-main, and indicator-diagrams from the pump. It is desirable that indicator-diagrams be obtained also from the steam-cylinders.

Observations of the length of the stroke should be made every five minutes; observations of the water-pressure gauges every fifteen minutes; observations of the remaining instruments—such as steam-gauge, vacuum-gauge, thermometer in pump-well, thermometer in feed-pipe, thermometers showing temperature of engine-room, boiler-room, and outside air, thermometer in flue, thermometer in steam-pipe if the boiler has steam heating surface, barometer and other instruments which may be used—every half-hour; indicator-diagrams should be

taken every half-hour, both from the steam and from the water cylinders. Should the diagrams from the pump be rectangular, they may be taken, if desired, with less frequency.

When the duty-trial embraces simply a test of the engine apart from the boiler, the course of procedure will be the same as that described, excepting that the fires will not be hauled and the special observations relating to the performance of the boiler will not be taken.

(5) In making preparation for the test, attention should be given to the following provisions in the arrangement of the apparatus:

The gauge attached to the force-main is liable to a considerable amount of fluctuation unless the gauge-cock is nearly closed. The practice of choking the cock is objectionable. The difficulty may be satisfactorily overcome and a nearly steady indication secured, with cock wide open, if a small reservoir having an air-chamber is interposed between the gauge and the force-main. By means of a gauge-glass on the side of the chamber and an air-valve, the average water-level may be adjusted to the height of the centre of the gauge, and correction for this element of variation avoided.

To determine the length of stroke in the case of direct-acting engines, a scale should be securely fastened to the frame which connects the steam and water cylinders, in a position parallel to the piston-rod, and a pointer attached to the rod so as to move back and forth over the graduations on the scale. The marks on the scale, which the pointer reaches at the two ends of the stroke, are thus readily observed and the distance moved over computed. If the length of the stroke can be determined by the use of some form of registering apparatus, this method of measurement is preferred. The personal errors in observing the exact scale-marks, which are liable to creep in, may thereby be avoided.

The form of calorimeter to be used for testing the quality of the steam is left to the decision of the person who conducts the trial. It is preferred that some form of continuous calorimeter be used which acts directly on the moisture tested. If

either the superheating calorimeter or the wire-drawing instrument be employed, the steam which it discharges is to be measured either by numerous short trials, made by condensing it in a barrel of water previously weighed, thereby obtaining the rate by which it is discharged, or by passing it through a surface-condenser of some simple construction, and measuring the whole quantity consumed. When neither of these instruments is at hand, and dependence must be placed upon the barrel calorimeter, scales should be used which are sensitive to a change in weight of a small fraction of a pound, and thermometers which may be read to tenths of a degree. The pipe which supplies the calorimeter should be thoroughly warmed and drained just previous to each test. In making the calculations, the specific heat of the material of the barrel should be taken into account, whether this be of metal or of wood.

If the steam is superheated, or if the boiler is provided with steam-heating surface, the temperature of the steam is to be taken by means of a high-grade thermometer resting in a cup holding oil or mercury, which is screwed into the steam-pipe so as to be surrounded by the current of steam. The temperature of the feed-water is preferably taken by means of a cup screwed into the feed-pipe in the same manner.

Indicator-pipes and connections used for the water-cylinders should be of ample size, and so far as possible free from bends: $\frac{3}{4}$-in. pipes are preferred, and the indicators should be attached one at each end of the cylinder. It should be remembered that indicator-springs which are correct under steam heat are erroneous when used for cold water. When steam springs are used, the amount of error should be determined if calculations are made of the indicated work done in the water-cylinders.

To avoid errors in conducting the test due to leakage of stop-valves either on the steam-pipes, feed-water pipes, or blow-off pipes, all these pipes not concerned in the operation of the plant under test should be disconnected.

(6) The engine is to be worked on the duty-trial, unless otherwise stipulated, at its rated capacity of discharge.

(7) In review of the method thus pointed out, the various steps may be summed up as follows:

a. Preliminary run to determine the temperature of the feed-water;

b. Erection of weighing apparatus, examination of pump, and test of plunger leakage;

c. Commencement of boiler-test;

d. " " engine-test;

e. Boiler and engine test go on simultaneously;

f. Close of engine-test;

g. " " boiler-test.

(8) It is desirable that the report of a duty-trial should be sufficiently full to show the performance of the engine and its various members in all other respects than the simple expression of the amount of duty performed. For this reason the horse-power developed by the steam-cylinders, the feed-water consumption per horse-power per hour, the steam accounted for by the indicator, and other information relating to the work of the engine in the capacity of a steam-engine, should be determined and given.

The efficiency of the mechanism of the engine should also be determined and stated, that is, the proportion which the work done upon the water bears to the work done in the steam-cylinders. This efficiency may be expressed by any formula in which the numerator is the duty and the denominator is the work done during the trial, measured from the indicator-cards taken from the steam-cylinders. This efficiency measures a quantity which is of primary importance in the operation of the engine and should always be carefully and exactly determined.

81. Fitting of the Engine for a Test, whether of efficiency or of capacity, is best done in advance of the trial, and ample time should be taken to see that not only all apparatus, but the engine itself, is in readiness; though, if the intention is, as is sometimes the case, simply to ascertain the condition of the engine as found, no other preparations are permissible than those customary before starting up. A good

example of the fitting up of a small high-speed engine is illustrated in Fig. 118, and, in plan, in Fig. 119, in which *AA* is the engine, *BB* its shaft, *CC* the indicators, *D* the indicator reducing-gear, driven from the crosshead; *EE* is the Prony Brake, receiving its cooling water at *F* and discharging it at *G*, and attached to the platform-scale arranged at *H*, the screw tightening its strap is at *I*. The speed-indicators were, in this case, of several kinds. Hand instruments of various kinds were used to check the records of

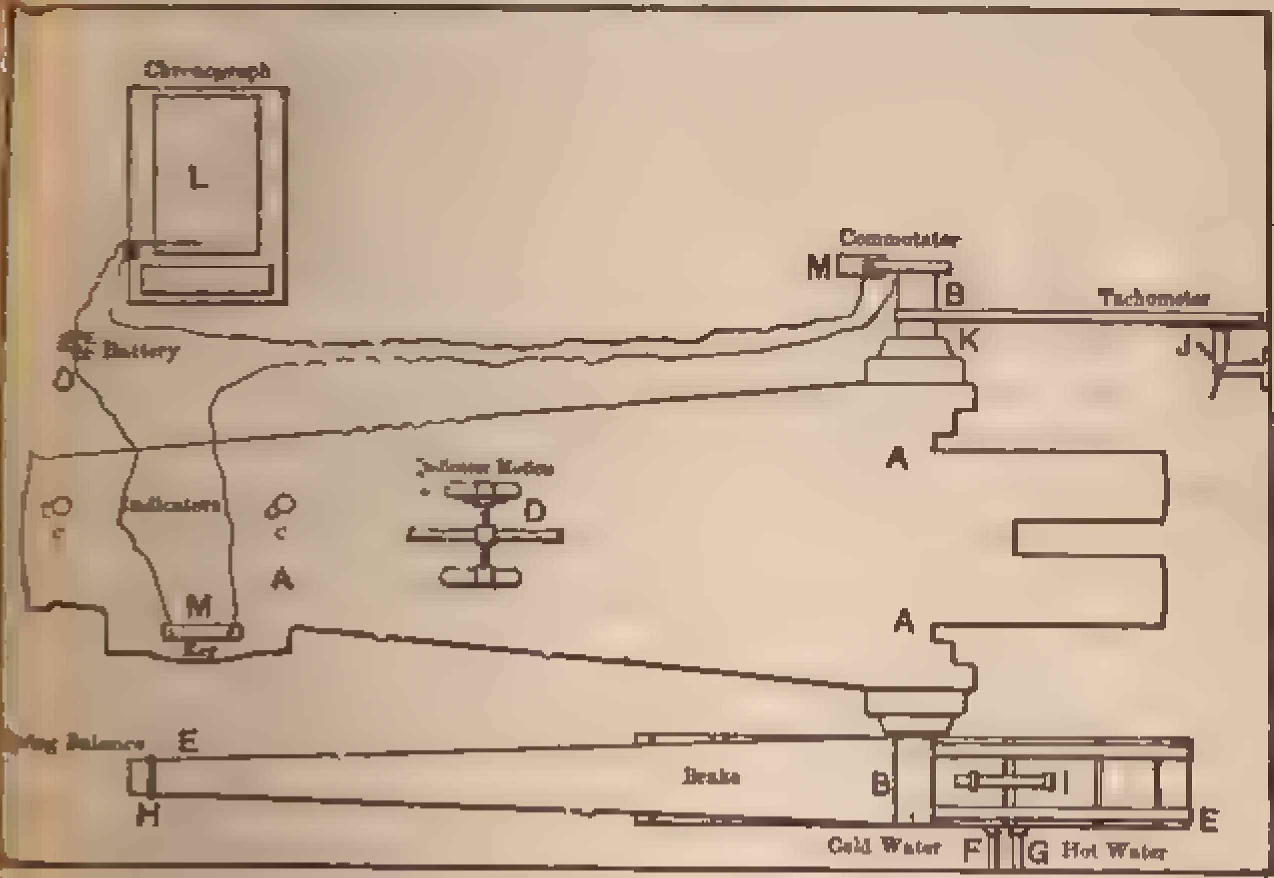


Fig. 119.—Fitting up the Engine.

the automatic instruments. A "tachometer," *J*, was attached and belted at *K* to the engine-shaft, and afforded a very convenient means of watching the momentary fluctuations due to variations of load, of steam-pressure, and of accidental disturbances. A chronograph at *L* was also attached, connected with a standard clock to beat seconds, and a current was derived from the battery at *O*. A commutator, *M*, was placed on the engine-shaft, making contact at each revolution, and a key, *N*, near the engine, for the purpose of breaking contact. A Brown

mercury speed-indicator served excellently well for a constant speed indicator. The chronograph was set in operation when the indicator-cards were taken, and thus gave the exact speed of the engine at that instant. Great care must be taken to keep the instruments, and the engine as well, in good order and well lubricated throughout the series of experiments.

82. Two Methods of Trial are available in testing steam-engines, both of which are found to be capable of giving exact results: (1) Measuring the energy supplied by the boiler in the form of heat transferred to the engine by the steam, and comparing the mechanical equivalent of this heat-energy with the quantity of mechanical energy obtained from the engine. (2) Determining the amount of energy rejected, as measured in the heat carried away by the exhaust, and similarly comparing this with the work done. In the first case, the quotient of the useful energy gained by the total energy expended is a measure of the efficiency of the system; in the second case, the same measure is obtained by dividing the work done by the sum of that quantity and the rejected energy. Of these two systems of trial, the first is that customarily employed by engineers for many years past; the second is that comparatively recently introduced by Messrs. Farey and Donkin. Both are fully described elsewhere.

The first system being adopted, the quantity of heat-energy expended is measured by determining the weight of steam produced and its physical condition, and the quantity of heat brought to the boiler by the feed water. The total heat communicated to the steam, less the heat received with the feed, is the net expenditure. It is usual to take a standard temperature at 0° F., 32° F., or 0° C., as that to which all temperature measurements are referred. In such case, assuming the standard point on the scale to be 0°, the total heat supplied by the boiler is ascertained by weighing the feed-water for a specified time, and thus determining the weight of steam, wet or dry, passing to the engine; next ascertaining what proportion of the fluid is still liquid, or what is the amount of superheating: computing the heat stored in the fluid; then, finally, deducting the

heat stored in the feed-water, both measured from 0°, thus obtaining the net quantity which comes from the fuel.

The second system being employed, the quantity of rejected heat is determined by measuring that received in the condenser and wasted in other ways. The total rejected heat consists of the following parts: (1) Heat carried away by air and vapor from the hot-well and by the water of condensation, measured from 0° or the standard point on the thermometer. (2) Heat received and carried away by the condensing water, the measurement being made between the limits of reception and rejection of that water. (3) Heat wasted by conduction and radiation from the exterior of the heated parts of the machine.

In illustration of such distribution of energy we find the following, as deduced by Prof. Ewing,* from data supplied by Mr. Main: †

Data.

Steam-pressure, absolute, lbs. per sq. in.	76
Time occupied by trial, hours	6
I. H. P.	127.4
Feed-water, lbs. per revolution (24 per min.)	1.394
Air-pump discharge, lbs. per revolution	51.1
Water drained from jackets, lbs. per revolution	0.186
Per cent. priming	4
Temperatures: feed, injection, and discharge	59°, 50°, 73°.4

Results.

Quality of steam	0.96
Quantity of steam supplied per revolution, lbs.	1.028
" " injection-water " " "	49.9
Latent heat of steam, B. T. U.	898
Heat in water of boiler, " (from 32° F.)	278°
" " " " feed, "	27
" " " " injection, B. T. U.	18
" " " " discharge, "	41.4

* Encyclopædia Britannica, 9th ed.; art. Steam-engine.

† Minutes Proc. Inst. C. E., vol. lxx.

Heat from boiler to engine, per revolution............	1377
" " " " jackets, " "	212
" " " total B. T. U. per revolution.........	1589
" returned to boiler, " " "	38
" *net* supply " " "	1551
" converted into work " " "	227
" total rejected " " "	1324

The loss by conduction and radiation, externally, was about 6 per cent. The actual efficiency of the engine was

$$E = \frac{227}{1551} = 0.146,$$

or not quite 15 per cent., while the thermodynamic efficiency was 0.335, more than twice as great.

This latter method is known as that of Messrs. Farey and Donkin.

In duty-trials of pumping-engines, the best system yet proposed is probably that already mentioned, which bases the efficiency determination upon the measured amount of work done by the system on a consumption of 1000000 B. T. U. supplied in the boiler-furnace, or used in the engine, as the case may be. The heat consumed should be taken to be all supplied by the fuel, or all received by the engine, including that wasted by all its accessories.

The useful work should be, wherever practicable, measured by the product of weight of water pumped, as ascertained by the use of a weir, into the head against which it is pumped, as measured by a pressure-gauge, or otherwise, at the pump-delivery. Losses by leakage, lost action, etc , are thus detected. Internal friction thus properly tells against the engine; external friction—in mains, etc.—is as properly ignored.

83. The Farey and Donkin System of trial of engines is one in which the quantity of heat supplied by the boiler and received by the engine is not directly determined, but is ascertained by observation of the quantities of heat rejected by the engine and carried away in the condensing water. This

method only applies to condensing-engines and to those which can be temporarily converted into condensing-engines for the purposes of the test. A boiler-trial is always a troublesome and disagreeable operation, and usually involves considerable expense both in preparation and in its conduct. Where it is only the engine that is to be tried and judged, the avoidance of a boiler-trial is a decided advantage. The ability to test an engine by itself is very often an important desideratum, and especially as permitting more frequent determinations of the condition of the machine and a more complete knowledge of its action at all times.

It has been seen that the heat supplied to any engine is disposed of in three directions: by conduction and radiation to surrounding objects; by conversion into mechanical work, and by rejection in the exhaust steam and the water accompanying it. Of these quantities the first is comparatively small, and is often entirely ignored as unimportant; the second ranges in good engines between, perhaps, 10 and 15 per cent., rarely exceeding the latter figure; while the last item includes, as a rule, above 85 per cent., and generally 90 per cent., of the total quantity sent over from the boilers. In the condensing-engine all this heat may be found and measured up in the water passing out at the delivery-pipe from the hot-well. It is obvious that the sum of the heat-equivalent of the indicated power of the engine, plus the heat so rejected, and the small quantity added to represent losses by radiation and conduction, will be the measure of the heat-supply from the boiler. To determine this total, therefore, we have but to measure the indicated power of the engine and the heat discharged from the condenser. The first of these processes is already understood. To secure the second measurement, it is only necessary to measure the flow of the heated water by a weir and notch, at the same time measuring its temperature by accurate thermometers. A high value for the quantity of heat discharged, per horse-power and per hour, indicates an inefficient engine; a low value is the proof of good economy.

The apparatus employed by Messrs. Farey and Donkin con-

sists, Fig. 120, of a simple measuring-box, *AA*, of convenient size, six or eight feet long usually, three to five feet wide and two to four feet deep, fitted with a notch, *D*, which is commonly about 6 inches wide. On the engraving this box is of iron, but it is perhaps oftenest of wood.*

It is fitted with transverse partitions *BB*, while at *D* a thin brass or copper plate has formed in it the notch producing the tumbling-bay. The notch in end of the box is larger than the notch in the plate, so that the approach of the water may not be interfered with. The box also is so placed that the water has a clear fall of 12 or 18 inches. The water from the hot-well is delivered into the box at one end, and flows over, under, and through the partitions, as shown, so as to be thoroughly mixed and the current steadied.

The box is provided at *C* with a standard fixed to the bottom, having a hole in it which receives the stem on which the float *e* moves loosely. At the top is a scale of inches capable of being adjusted by screws, while the float carries a pointer *f* which moves up and down this scale with the float.

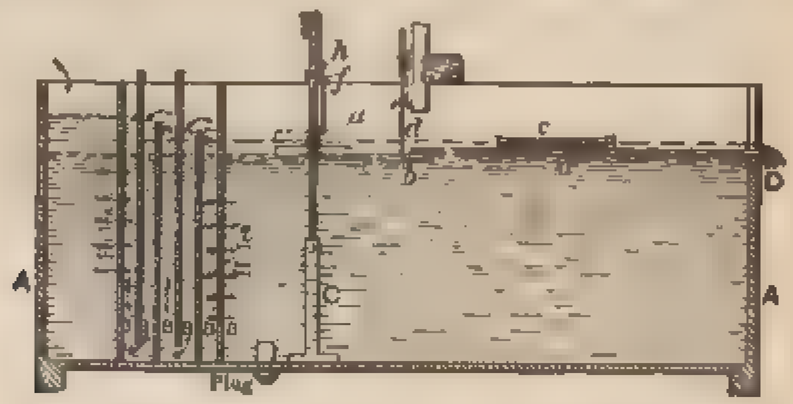


FIG. 120.—FAREY AND DONKIN'S APPARATUS.

To fix the zero-point, a straight-edge *a* is provided, and another, *b*, forming an extension of the bottom edge of *a*; *a* is then placed with one end resting on the notch, while *b* is beneath the gauge *d*, this gauge being free to move vertically in its holder; *a* is then adjusted until the spirit-level *c* shows it to be level when the gauge *d* is fixed by its clamping-screw. The straight-edge and spirit-level are then removed. A scale can

* London Engineering, Feb. 5, 1875.

now be fixed so that its zero agrees with a mark on the gauge stem. If a float is to be employed, the gauge is only used to determine the zero of the float-scale. The float having been put in place, water is admitted into the box until the surface is found just to touch the point of the point-gauge, and the scale is adjusted so that the zero-point agrees with the index of the float.

The depth of water in the notch being measured by the gauge or the float, and the width of the notch being exactly known, the quantity of water flowing is at once readily computed by use of the standard formulas for flow, through a notch or over a weir. The temperatures of the water being taken at the same time, before the water enters and as it leaves the condenser, the product of the mean weight of water flowing per hour by the mean range of temperature measures the heat-units discharged. This quantity, divided by the mean indicator horse-power for the same period, gives the desired figures per indicator horse-power per hour, or

$$H' = \frac{V \times D \times S \times (T_1 - T_2)}{I. H. P.};$$

when H' = heat-units as above;
V = volume of water flowing per hour;
D = density of water at the observed temperature;
S = specific heat, usually taken as unity;
T = observed temperature;
$I. H. P.$ = indicated horse-power;

The quantity H' is often called the constant for the engine. Since each horse-power demands

$$\frac{1980000}{772} = 2564.5, \quad \text{or} \quad \frac{33000}{772} = 42.75,$$

heat-units per hour or per minute, the quantity of steam supplying the heat converted into work per hour is

$$\frac{2564.5}{h - t} = w,$$

when h and t are the total heat of the steam and the temperature of the condenser, ranging from 2½ to 2¼ pounds, according to circumstances. The heat discharged, H', being given, the weight of steam supplying it is

$$\frac{H'}{h-t} = w',$$

varying from about 15 pounds upward per *I. H. P.*, the total of both items thus measuring the demand on the boiler, amounting to $w + w' = 17$ or more pounds per horse-power and per hour in good engines. If the boiler "foams" or "primes," this expenditure of feed-water is correspondingly increased. This relation being established, any variation in it or in the "constant" for the engine, as shown at the "tumbling-bay" or weir-notch, indicates some change in the working of the engine, and will call for attention.

Mr. Donkin gives the following table for use in making trials by the Farey and Donkin method:

WEIGHT OF WATER THAT WILL FLOW OVER A TUMBLING-BAY SIX INCHES WIDE.

Inches over Bay.	Pounds of Water per Minute	Inches over Bay	Pounds of Water per Minute.	Inches over Bay.	Pounds of Water per Minute.	Inches over Bay.	Pounds of Water per Minute.
1½	274	2⅜	547	3¼	874	4⅛	1250
1 9/16	292	2 7/16	568	3 5/16	900	4 3/16	1279
1⅝	310	2½	589	3⅜	926	4¼	1306
1 11/16	327	2 9/16	612	3 7/16	951	4 5/16	1336
1¾	345	2⅝	634	3½	977	4⅜	1365
1 13/16	365	2 11/16	657	3 9/16	1003	4 7/16	1394
1⅞	383	2¾	680	3⅝	1030	4½	1424
1 15/16	402	2 13/16	704	3 11/16	1056	4 9/16	1454
2	421	2⅞	727	3¾	1083	4⅝	1483
2 1/16	442	2 15/16	751	3 13/16	1112	4 11/16	1514
2⅛	462	3	775	3⅞	1139	4¾	1544
2 3/16	483	3 1/16	800	3 15/16	1166	4 13/16	1575
2¼	503	3⅛	825	4	1193	4⅞	1605
2 5/16	525	3 3/16	850	4 1/16	1221	4 15/16	1635

N. B.—10 pounds of water is taken equal to one gallon.

84. Trials of Gas-Engines [illegible] tion [illegible] quantity of gas consumed [illegible] also, if satisfactorily [illegible] the several wastes [illegible] radiation within the working [illegible] The volume of water flowing [illegible] tion of temperature [illegible] but the measurement [illegible] This involves the measurement [illegible] the gases entering and leaving the engine [illegible] of temperature. [illegible] quantity of [illegible] and the gas entering the working [illegible] not certain in any case that [illegible] tained by the measurement [illegible] nary work, it may be [illegible] tion to accuracy. The [illegible] ly ascertained by analysis. [illegible] sizes exceeding ten [illegible] above 20 cubic feet [illegible] hour, or 30 feet [illegible] minating gas often may give a result less satisfactory by ten per cent. or more. Many gas-engines of less perfect construction demand double this quantity and upward. A cubic foot of good gas should supply about 600 B. T. U. of heat; a cubic metre should yield about 5600 calories. The theoretical mixture will be usually found to be not far from seven volumes of air to one of gas; but it is better to use a slight excess of air.

As has been already seen, the complete indicator-card is, in the compression type of engine, only obtained after four revolutions, and the observer should be careful to see that he has the diagram of the complete cycle before removing the pencil from the paper.

The data are complete when they permit the computer to show precisely how much gas, how much heat, and how much

energy are supplied to the engine; how much is applied usefully; how much is wasted and what are the measures in detail of all wastes; securing results in such manner, that it is made possible to construct an account that shall exactly or approximately exhibit on its balance-sheet all receipts and expenditures and an exact balance.

85. **Simple and Binary Vapor-engine Trials** involve no peculiar methods or operations. In these, as in all other cases, the problem of the engineer is the determination of the heat-energy developed and applied in the engine and of the nature and magnitudes of all wastes. In vapor-engines, as in the ammonia or the naphtha-vapor engine, the only differences in its working, when compared with the steam-engine, are due to peculiarities of physical properties, and involve no essential modification of the method of heat or power measurement. The purpose of the trial is commonly to obtain a comparison of efficiency with that obtainable under similar circumstances, with a steam-engine of equally good design and construction, or of standard make and operating in the customary manner. A common practice, on the part of promoters of new schemes in this direction, is to exhibit a comparison with a comparatively wasteful and badly constructed steam-engine. The engineer making such trials should be especially careful in this matter.

The Binary Vapor-engine is commonly a complex machine, composed of a steam-engine and a simple vapor-engine utilizing the heat of the rejected steam. This combination is tested with the steam-engine of fairly comparable design and construction. As a rule, the comparison lies between the combined motor and a condensing steam-engine. The trial determines the efficiency of the steam-engine and the vapor-engine, separately and combined, and should give complete data relating to quantities of heat transferred and transformed, of fuel, and of fluids employed, and of work, useful and lost, as well as of the power developed by each. When testing other vapors than steam, it is often important that their essential chemical and physical characteristics should be redetermined for the occasion; as they sometimes vary somewhat, as in the case of the petroleum

vapors, and they may differ from the recorded data of the treatises taken as authority.

86. Gas and Vapor Engine Trials may thus demand special treatment in some cases. The specific heats of gas-mixtures may require to be determined; the specific heats of vapors may be undetermined, or their recorded values may be inaccessible. In such cases it may become the duty of the engineer to ascertain their values by computation or by direct experiment.

For example: In a trial of a gas-engine by Messrs. Brooks and Steward it was necessary, in order to determine the quantity of heat stored and wasted in the exhaust gases, to determine the specific heat of the mixture of steam, carbon dioxide, and nitrogen thus: * The analysis of the gas used in the tests is—

		By volume.
H	Hydrogen,	.395
CH_4	Marsh-gas,	.373
N	Nitrogen,	.082
C_2H_4, etc.	Heavy hydrocarbons,	.066
CO	Carbonic oxide,	.043
O	Oxygen,	.014
H_2O_1, CO_2, H_2S, etc.,	Water-vapor, impurities, etc.,	.027
		1.000

By weight its composition is found to be—

	Cu. meters.		Densities.†		Kilos per cu. m.	W't p. unit.
H	.395	×	.087	=	.035	.058
CH_4	.373	×	.694	=	.258	.426
N	.082	×	1.215	=	.099	.163
C_2H_4, etc.	.066	×	1.84	=	.121	.200
CO	.043	×	1.215	=	.052	.086
O	.014	×	1.388	=	.019	.031
H_2O_1, etc.	.027	×	$\sim$.8	=	.022	.036
	1.000	×	.606	=	.606	1.000

* Van Nostrand's Magazine, 1883.

† Schöttler: Die Gasmaschine, p. 77.

By "density" is meant the weight of one cubic meter in kilogrammes. One cubic meter of the gas in question weighs 0.606 kilos.

Upon complete combustion the gas develops heat per cubic meter as follows:

	Calories.*				Calories.
From H	29060	×	.035	=	1020
" CH_4	11710	×	.258	=	3020
" C_2H_4, etc.	11000	×	.121	=	1330
" CO	2400	×	.052	=	125
				per cu. m.	5495

and per kilog. gas $\frac{5495}{.606} = 9070$ calories.

In British measures, one cubic foot of gas develops 617.5 heat-units.

To determine the amount of air supplied for complete combustion, it is necessary to ascertain the quantity of oxygen in chemical combination with the combustible constituents of the gas.

	$2H + O$	$= H_2O$	
by volume	$2 + 1$	$= 2$	
by weight	$2 + 16$	$= 18$	
	$CH_4 + 4O$	$= CO_2$	$+ 2H_2O$
by volume	$2 + 4$	$= 2$	$+ 4$
by weight	$16 + 64$	$= 44$	$+ 36$
	$C_3H_6 + 9O$	$= 3CO_2$	$+ 3H_2O$
by volume	$2 + 9$	$= 6$	$+ 6$
by weight	$42 + 144$	$= 132$	$+ 54$
	$CO + O$	$= CO_2$	
by volume	$2 + 1$	$= 2$	
by weight	$28 + 16$	$= 44$	

The combining proportions are—

By volume

$$1H + \tfrac{1}{2}O = 1H_2O$$
$$1CH_4 + 2O = 1CO_2 + 2H_2O$$
$$1C_3H_6 + 4\tfrac{1}{2}O = 3CO_2 + 3H_2O$$
$$1CO + \tfrac{1}{2}O = 1CO_2$$

* Schöttler. Die Gasmaschine, p. 80.

By weight—

$$
\begin{array}{llll}
1H & + \; 8O & = & 9H_2O \\
1CH_4 & + \; 4O & = & \frac{11}{4}CO_2 + \frac{9}{4}H_2O \\
1C_2H_6 & + \frac{24}{7}O & = & \frac{22}{7}CO_2 + \frac{9}{7}H_2O \\
1CO & + \; \frac{4}{7}O & = & \frac{11}{7}CO_2
\end{array}
$$

The volume of oxygen required for the combustion of 1 volume of gas is—

$$
\begin{array}{llcrcr}
H & .395 & \times & \frac{1}{2} & = & .197 \\
CH_4 & .373 & \times & 2 & = & .746 \\
C_2H_6 & .066 & \times & 4\frac{1}{2} & = & .297 \\
CO & .043 & \times & \frac{1}{2} & = & .022 \\
\hline
 & & & & & 1.262
\end{array}
$$

$$
\begin{array}{lr}
\text{Less O in gas .014} \quad . \; . \; . \; . & .014 \\
\hline
 & 1.248
\end{array}
$$

Taking oxygen as 21 per cent. in atmospheric air, the volume of air needed is

$$\frac{1.248}{.21} = 5.94 \text{ per volume gas.}$$

Since air weighs 1.251 kilos per cu. meter, the ratio by weight is

$$\frac{5.94 \times 1.251}{1 \times .606} = 12.26 \text{ air to gas } 1.$$

From the combustion of 1 unit weight of gas with 12.26 air, there results 13.26 units weight of a mixture the composition of which will be—

$$
\begin{array}{ll}
CO_2 & \left\{\begin{array}{llcrcr}
(CH_4) & .426 & \times & \frac{11}{4} & = & 1.171 \\
(C_2H_6) & .200 & \times & \frac{22}{7} & = & .629 \\
(CO) & .086 & \times & \frac{11}{7} & = & .135
\end{array}\right\} \quad 1.93 \\
H_2O & \left\{\begin{array}{llcrcr}
(H) & .058 & \times & 9 & = & .522 \\
(CH_4) & .426 & \times & \frac{9}{4} & = & .958 \\
(C_2H_6) & .200 & \times & \frac{9}{7} & = & .257
\end{array}\right\} \quad 1.74 \\
N & \left\{\begin{array}{lr}
\text{from the air,} \; . \; . & 9.407 \\
\text{in gas itself,} \; . \; . & .163
\end{array}\right\} \quad 6.57
\end{array}
$$

$$
\begin{array}{lr}
\text{Impurities in gas,} \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . & 0.03 \\
\hline
 & 13.27
\end{array}
$$

Per unit weight of mixture the composition will be—

CO_2	.146
H_2O	.131
N	.721
Impurities, ...	.002
	1.000

The volume which 13.27 kilos of products of combustion will occupy is found from the known volumes of the constituent gases as follows:

	Kilos.		Cu. m. per kilo.		Cu. m.
CO_2	1.93	×	.524	=	1.011
H_2O	1.74	×	1.28	=	2.227
N	9.57	×	.823	=	7.876
Impurities, .	.03	×	~.9	=	.027
					11.141

The products of combustion occupy 11.141 cu. m. to each kilog. of gas. To find the ratio per cu. meter of gas, we have simply to multiply by 0.606, the number of kilos in a cubic meter, and get 6.751. As 6.94 cu. m. of air and gas are needed to every cu. m. gas, by a contraction of 2.7 per cent. combustion takes place.

The specific heats of the products of combustion are determined from the specific heats of the several component gases as follows:

Specific heat at constant pressure (water = 1):

$$C_p = \left\{ \begin{array}{ll} .2169 \times .146\ (CO_2) & = .0317 \\ .4805 \times .131\ (H_2O) & = .0629 \\ .2438 \times .721\ (N) & = .1758 \\ \sim .4 \times .002\ (\text{impurities}) & = .0008 \end{array} \right\} = 0.2712.$$

Specific heat at constant volume (water = 1):

$$C_v = \left\{ \begin{array}{ll} .1714 \times .146\ (CO_2) & = .0250 \\ .3694 \times .131\ (H_2O) & = .0484 \\ .1727 \times .721\ (N) & = .1245 \\ \sim .3 \times .002\ (\text{impurities}) & = .0006 \end{array} \right\} = 0.1985.$$

The ratio of these specific heats is the exponent of adiabatic expansion, and is found to be

$$\gamma = \frac{C_p}{C_v} = \frac{0.2712}{0.1985} = 1.366.$$

Since there is always an excess of air present, these values will be somewhat modified by that fact. From the meter records the ratio of air to gas by volume was found to be 6.63 to 1; by weight the ratio is

$$\frac{6.63 \times 1.251}{1 \times .606} = 13.68.$$

Since for complete combustion only 12.26 parts of air by weight are needed, there are 1.42 parts in excess. The specific heats of air being $C_p = .2375$ and $C_v = .1684$, the effect of the excess of air will be to reduce the specific heat slightly.

$$C_p = \frac{(.2712 \times 13.26) + (.2375 \times 1.42)}{14.68} = .268;$$

$$C_v = \frac{(.1985 \times 13.26) + (.1684 \times 1.42)}{14.68} = .196;$$

$$\gamma = \frac{C_p}{C_v} = \frac{.268}{.196} = 1.37.$$

87. **The Scheme of the Trial** should be carefully prepared in advance, and should be so planned as to secure the needed data with certainty and accuracy. The first consideration is the purpose of the proposed trial, and the first work done the arrangement of a general plan that shall enable the observers to collect with accuracy and certainty all the needed data, and to record them conveniently and in most available form. The next matter to be studied is the reduction of all general and special operations of the trial to a complete and efficient system, in which every part shall be made as far as possible contributory to the efficiency and fruitfulness of every

other part; in which each observer shall be so stationed and so instructed that he may secure the data assigned him for collection with least difficulty, risk, and uncertainty, and shall have his own work checked, and shall aid in checking the work of others, as completely as possible. No essential data should remain unchecked, and every subsequent calculation based upon them should also be made by at least two computers independently.

The plan of the work being settled upon, each detail should be studied by itself, and every provision that experience and foresight can suggest should be taken to insure perfection of the scheme. A preliminary and informal trial will then be likely to reveal any serious defect, which being corrected, the final and official trial may be fully expected to give thoroughly reliable results.

88. Competitive Trials of Engines are sometimes conducted by the engineer, either to determine which of two or more competing forms of engine is to be accepted by the purchaser, or by his client, or, as at exhibitions of various kinds, simply to ascertain the power and efficiency of two or more engines, with a view to deciding their relative merits as types of engine, or as representing the best practice of their builders. It is largely through this kind of competition that the best known systems of standard engine-trial have been developed. Examples of such are illustrated by the following regulations, adopted at the exhibitions of the Franklin Institute of the State of Pennsylvania:

NOTICE: –Exhibitors of engines, desiring quantitative tests made of their exhibits, must make formal application for such tests in advance.

Engines can be exhibited, but will not be tested unless formal application and agreement to the following code are completed within the specified time.

Parties desiring tests made of their engines can have them made by making formal application therefor, and by subscribing to and fulfilling the conditions of the Code.

All tests will be quantitative, and will, once begun, not be abridged, save by special agreement with the judges.

Tests of regularity of speed, however, will be made independently of other measurements.

The Committee reserves the right to limit the number of engines tested and to elect which engines shall be tested, if time will not permit complete tests for all making formal application.

Competitive tests will not be made, save on the joint application of the two or more parties desiring them, who must, previous to the tests, agree on the rating of the various points of the engine (see Article 9), and subscribe to the Code, agreeing to abide by the decision of the judges without appeal.

Conditions of Exhibition and Test.

(1) The cylinders of the engines entered may be of any capacity and proportion of stroke to diameter.

(2) Each cylinder shall be drilled and tapped by the builder, for indicator connections, by means of one-half ($\frac{1}{2}$) inch pipe in the usual manner, and to the satisfaction of the judges. Pet drainage-cocks must be on the cylinder. The cross-head or some other point must be drilled for the indicator-cord attachment.

(3) Each cylinder shall be drilled and plugged at both ends so as to admit of being completely filled with water and emptied by means of a one-half ($\frac{1}{2}$) inch pipe, in order to determine the clearance and the piston-displacement of one stroke at each end. These data will be obtained both hot and cold

(4) The steam and exhaust valves will be tested under full steam-pressure, ninety (90) pounds per square inch by the gauge, unless some other pressure has been agreed upon for the test.

(5) The tightness of the piston-packing will be determined by removing the back cylinder-head and subjecting the piston to full boiler-pressure on each centre.

(6) Each maker is requested to use such diameter of band

fly-wheel or of pulley as shall give a belt-speed of 4000 feet per minute. Should he require a different belt-speed, he will specially note the same, in communicating with the Committee.

(7) Each exhibitor will be required to furnish his own connections with the main steam-pipe, the main injection-pipe, and the main overflow pipe or tanks.

(8) Each exhibitor will be furnished with space at the regular rates established for the exhibition, in which space he must build his foundations at his own cost, and subject to the approval of the Superintendent.

(9) Each exhibitor will communicate to the chairman of the Committee such a description and drawings of the engine exhibited as will facilitate the labors of that Committee, together with his claims as to meritorious points for his exhibit.

The following points will have special consideration:*

1. Economy of steam.
2. Regularity of speed.
3. Concentration of power.
4. Durability of construction.
5. Simplicity of design.
6. Perfection of proportions.
7. Finish of parts.

Each exhibitor must file the following data, before the tests, viz.:

Diameter of steam-cylinder to nearest hundredth of an inch.
Diameter of piston-rod " " "
Diameter of steam-pipe " " "
Diameter of exhaust-pipe " " "
Diameter of fly-wheel " " "
Width of the face of fly-wheel " " "
Weight of fly-wheel in pounds.
Area of steam-ports, each to nearest hundredth of an inch.
Area of exhaust-ports, " " " "
Stroke of engine, " " " "
Indicated horse-power of engine when believed to be working most economically.
Revolutions of crank per minute.
Weight of whole engine, exclusive only of fly-wheel.

* These are the points referred to in the special notice concerning the value of which agreement must be had previous to the competitive tests.

When a condenser is used and its air-pump driven by the engine, the following additional data will be required, viz.:

Diameter of air-pumps to nearest one-hundredth of an inch,
Diameter of injection-pipe " " "
Diameter of overflow-pipe " " "
Stroke of air-pump piston " " "

And if an independent condenser is used, *i.e.*, not driven by the engine, give

Diameter of injection-pipe to nearest one-hundredth of an inch,
Diameter of overflow-pipe " " " "
Drawings of condenser used, any other data peculiar to it, and a full description.

Preparations for the Tests.

(10) The steam for the tests will be furnished by the exhibition-boilers, and will come from boilers specially set apart for the purpose of the tests. It will be charged for at regular rates of three (3) cents per indicated horse-power per hour. Steam will be furnished to exhibitors one week before the tests are made, if desired.

No charge will be made for the services of attendants or experts, or the use of apparatus, unless in some extraordinary case, when the cost will be fixed by the superintendent.

(11) The steam-pressure used will be subject to the wish of the exhibitor, but shall not exceed ninety (90) pounds per square inch, by the gauge.

A special standard gauge will be used during the tests, and subjected to careful tests before and after use.

(12) The safety-valve will be set to blow off at ten (10) pounds above the pressure fixed upon.

(13) The thermal value, the temperature, and the pressure will be taken by means of scale-calorimeters, thermometers, and standard gauges at the boiler, at the steam-chest, and at the exhaust, if the engine is non-condensing.

The thermometers, calorimeters, etc., will be furnished by the exhibition, but the exhibitor must do such mechanical work, must furnish such piping, tools and materials, as are neces-

sary to make the required attachments, at his own cost, and subject to the orders of the Committee.

(14) The temperatures of injection and of hot-well will be taken with standard thermometers, in the case of condensing-engines.

(15) The water used will be taken from the city mains.

The feed-water for the boilers will be weighed by means of scales and a large tank, and will be run into a smaller supplemental tank, from which it will be pumped into the test-boilers by means of a feed-pump actuated by steam from other boilers.

The condensing water used will, in the case of condensing-engines, be measured after leaving the hot-well, in two carefully gauged tanks, alternately filled and emptied, the temperature also being taken.

The known weight of steam used will be subtracted from the overflow.

The injection-water will be weighed in large tanks, and its temperature taken.

The injection-water will not be delivered under pressure.

(16) The number of revolutions of the engines will be taken by a continuous counter attached to the crank-shaft.

The variations in speed for one minute will be taken at each quarter of an hour by means of an electric chronograph, connected with a standard clock, beating seconds.

The variations in speed during one stroke will be taken by an acoustic chronograph at fifteen minutes' intervals.

Special tests of speed alone, under varying loads, will be made if desired, and close attention will be had to this point in all cases.

(17) A standard barometer and thermometer will be read at fifteen-minute intervals during the trial.

(18) The vacuum of condensing-engines will be read by a gauge, carefully compared before and after the trials.

(19) All of the gauges, indicators, and thermometers used shall be carefully tested before and after the trials, and the

party whose engine is tested shall have the right to be present in person or by agent at these tests.

(20) The indicator-diagrams will be taken at fifteen (15) minute intervals, and will be read for—

Initial pressure,
Pressure at cut-off,
Terminal pressure,
Counter-pressure at mid-stroke,
Maximum compression-pressure,
Mean effective pressure,
Point of cut-off,
Release of steam,
Exhaust closure.

From the diagrams will be computed the indicated steam at the point of cut-off and at release, as also the actual steam from boilers per horse-power per hour.

(21) The Committee will test the engine at the load desired by the exhibitor of it, unless circumstances shall render it impossible to meet his wishes.

If the load does not exceed seventy-five (75) indicated horse-power, the net load will be measured by a transmitting dynamometer.

(22) At the close of the regular trial the engine will have its belt taken off, and be run for one hour for friction-diagrams.

(23) Unless otherwise arranged, the trials will last ten (10) hours.

(24) No account will be taken of the coal burned, but the economy of the engine will be deduced from the actual steam used and water weighed to the boiler.

The trial will begin with the established pressure.

The level of the water in the boiler and the pressure of the steam will be kept as nearly constant as possible during the whole of the trial.

The whole weight of the water fed to the boiler, subject to proper deductions for waste, and to corrections for variation of level in the boiler, will be multiplied by its thermal value as steam at the steam-chest, and divided by the product of the

indicated horse-power of the engine, and the number of hours of the test.

The resulting quotient will be used to divide twenty-five hundred and fifty-seven and sixty-nine one-hundredths (2557.69) British thermal units, * giving the efficiency of the engine as compared with the mechanical equivalent of the heat furnished to it, and therefore its efficiency, as a means of converting heat into work.

The net horse-power of the engine will be used for computation similarly to the indicated horse-power, and the result will be taken as the measure of the efficiency of the engine, both as a means of converting heat into work, and as a machine for the transmission of power.

This latter shall be considered the true measure of the efficiency of the engine.

89. Regulations for Competitive Boiler-trials are illustrated by the following, adopted at the same time as the preceding:

NOTICE.—Boilers may be exhibited and used at the Exhibition, but quantitative tests of their efficiency will not be made except upon formal application, and the acceptance of the subjoined code.

Competitive tests will not be made unless at the joint request of the parties desiring them and until such parties have agreed to and subscribed to this code, and fixed upon a rating for the points enumerated in Article 4.

The Committee reserve the right to limit the number of tests made, should time and opportunity not permit the completion of all the tests desired.

Preliminaries to the Tests.

(1) *Capacity.*—The boilers entered may be of any capacity having an evaporative power not less than seven hundred and fifty (750) pounds of water per hour.

Each boiler must be so drilled and fitted with proper pipes and cocks that the judges may be enabled to determine readily

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* Joule's Equivalent is here taken as 774.1 foot-pounds.

its whole water capacity by filling and emptying the boiler and weighing the contents.

(2) *Pipes and Valves.*—Each exhibitor will furnish all the pipes and valves necessary to make connection with the main water and steam pipes in a proper manner, and subject to the orders of the Superintendent.

He will also make any alterations in water and steam pipes required for the tests, furnishing all tools, piping, cocks, and mechanical labor at his own cost.

(3) *Space.*—Each exhibitor will be furnished with space at the regular rates established for the Exhibition, in which space he must build his foundations and boiler-setting, and make connection with the chimney-flue, if required, at his own cost, and subject to the approval of the Superintendent.

(4) *Specifications.*—Each exhibitor must furnish to the Chairman of the Committee such description and drawing both of the boiler in position and of the details of the boiler as will facilitate the labor of that Committee, together with his claims as to the meritorious points of his exhibit.

The following points will have special consideration:

1. Economy of fuel; 2. Economy of material and labor of construction; 3. Evaporative power; space occupied; 4. Simplicity and accessibility of parts; 5. Durability of whole structure.

Exhibitors desiring a competitive test made must agree upon a rating for these points before it will be made.

Exhibitors must also file the following data:

Area of heating-surface to the nearest hundredth of a foot; area of grate-surface to the nearest hundredth of a foot; area of calorimeter-surface to the nearest hundredth of a foot; area of chimney-flue surface to the nearest hundredth of a foot; height of chimney desired; number of pounds of coal per square foot of grate to be burned per hour.

Should the determinations of these preliminaries by the Committee differ in result from those of the exhibitor, he will be required to give all the details of his calculations, and an agreement must be reached before proceeding with the test.

Preparations for the Tests.

(5) *Coal.*—Anthracite coal will be used and will be furnished free of charge, provided the steam made is used for the general purposes of the Exhibition.

The same quality and size of coal will be used in all the tests, unless special arrangements be made for another kind of fuel.

An analysis will be made of the coal used.

The coal will be weighed to the boiler.

(6) *Water.*—The water used will be taken from the city mains.

The feed-water for the boilers will be weighed by means of scales and a large tank, and will be run into a smaller supplemental tank, from which it will be pumped into the boiler by means of a feed-pump actuated by steam from the boilers.

The temperature of the feed-water will be taken by means of a standard thermometer in the supplemental tank.

(7) *Pressure.*—The steam-pressure used shall not exceed ninety (90) pounds per square inch by the gauge, unless by special arrangement with the Committee.

A standard gauge will be used, and also a standard thermometer immersed in a mercury-pocket in the steam-space.

(8) *Safety-valve.*—The safety-valve will be set to blow off at ten (10) pounds above the pressure fixed upon.

(9) *Leaks.*—Within twenty-four (24) hours preceding the test of the boiler it must be subjected to hydraulic pressure ten (10) pounds greater than its steam-pressure during the test, and proved to be perfectly tight.

(10) *Attendants.*—The attendants in charge of the boiler tested must be approved by the party whose boiler is tested and by the judges. All attendants are to be subject to the orders of the judges during the progress of the test.

(11) *Ashes.*—All ashes will be weighed on being withdrawn from the ash-pit, and must not be damped until weighed.

(12) *Calorimeters.*—The calorimeters used will consist of a barrel, scale, and hand thermometer.

Two calorimeters will be used and simultaneous observations made at fifteen (15) minute intervals.

(13) *Fires.*—The exhibitor shall be allowed one day previous to the test to clean boilers and grates.

The steam having reached the required pressure, the ash-pit shall be thoroughly cleaned and swept, and thereafter the fire maintained as nearly uniform as possible, the test closing with the same depth and intensity of fire as it opened.

This point is to be decided by the judges, who may make allowances if it be clearly shown to have been impossible to maintain uniform fires.

If in the judgment of the Committee the firing is inefficiently or improperly done, the test may be terminated at any time, and a repetition of the test granted or refused.

(14) *Pyrometer.*—The temperature of the gases of combustion immediately upon entering the chimney-flue shall be taken by means of a suitable pyrometer, read at fifteen (15) minute intervals, and close to the boiler.

(15) *Manometer, Barometer.*—The vacuum in the chimney-flue shall be taken by means of a water-manometer, read at fifteen (15) minute intervals, if natural draught is used.

If a forced blast is used the manometer will be placed on the conduit to the ash-pit.

A barometer will be read simultaneously.

(16) *Duration.*—Unless otherwise arranged, the tests will last ten (10) hours.

(17) *Economy and Efficiency of the Boiler.*—The level of the water in the boiler and the state of the fire must be kept as nearly constant as possible during the whole of the trial.

The weight of the water in the boiler for each one-quarter of an inch, on the glass water-gauge, will be carefully determined and recorded previous to the test, and proper correction for unavoidable changes of level made.

The weight of water fed to the boiler, subject to proper corrections, will be multiplied by its observed thermal value as steam.

From this product thermal units of heat brought in by the feed will be subtracted.

The remainder will be divided by nine hundred and sixty-five and seven-tenths (965.7) British thermal units,* giving the number of pounds of water evaporated from and at 212 degrees Fahrenheit.

This latter quantity will be divided by the weight of coal burned, less weight of dry ashes, giving the number of pounds of water evaporated per pound of combustible.

This shall be taken as the measure of the efficiency of the boiler.

(18) *Nominal Horse-power.*—The nominal horse-power of the boiler will be deduced by dividing the number of pounds of water evaporated from and at 212 degrees Fahrenheit per hour by 30.

(19) *Evaporative Power.*—The evaporative power of the boiler will be determined by dividing the nominal horse-power of the boiler by the number of cubic feet of space it occupies.

The space occupied by a boiler and its appurtenances will be regarded as the product of the square feet of floor-space occupied by its extreme height in feet.

Steam-pump tests have been conducted at such exhibitions under the following regulations, written, originally, by Mr. Hill:

Regulations for Test of Steam Pumps.

(1) Steam will be furnished by the boilers used in the experiments upon automatic and slide-valve engines; the pressure will be taken in the pipe as near the stop-valve as convenient. The pressure in the boilers will be maintained as uniformly as possible at (75) seventy-five pounds per sq. inch above atmosphere.

(2) A calorimeter test of the quality of steam furnished will be made regularly every thirty (30) minutes. The steam-pipe will be tapped in the last horizontal joint toward the pumps for calorimeter connection.

* The value taken here for the latent heat of steam at the boiling-point.

(3) The exhaust will be delivered to a surface-condenser having not less than 500 sq. feet of condensing surface; the condensing water will be obtained from the city mains; water of condensation will be collected in a tank placed under the outlet nozzle of condenser.

(4) The suction tank will be placed below the level of pump; the distance from bottom of tank to centre of water-cylinder will be uniform. For all contestants the vertical head of water in suction-tank will be taken with a sliding-hook gauge, at regular intervals.

(5) The delivery-tank will be placed on a staging directly over the water-cylinder of pump, the discharge opening of water-cylinder will be connected with a 6-inch vertical stand-pipe furnished with a direct-weighted safety-valve, the height of stand-pipe from centre of water-cylinder to centre of orifice of safety-valve will be 10 feet. The safety-valve will be loaded to create a resistance per sq. inch equivalent to a dynamic head of 150 feet less the height of stand-pipe (10) feet.

(6) The measuring-tank will be placed (vertically) between suction-tank and delivery-tank; the measuring tank will be divided by a vertical partition in the centre into two compartments. Each compartment will have a capacity of 300 cubic feet; the water will be delivered from the receiving-tank into the measuring-tank through a (6) six-inch swinging nozzle. The nozzle will be directed over one tank until it has been filled and the water breaks over the dividing partition, when it will be swung over the empty tank; in the mean time the temperature of the water in the full tank will be noted, the number of tank entered in the log, and the contents drawn off through an (8) eight-inch delivery-pipe into the suction-tank below; this operation will be repeated regularly during the run. The precise capacity of each compartment of the receiving-tank will be determined prior to the experiments by filling each to the crest of partition, and drawing off, weighing, and noting the temperature of contents.

(7) The duration of run will be fixed at (5) five hours. Previous to the commencement of run, the steam will be turned

on and the pump will be operated until all parts have acquired the working temperature.

(8) The pressure of atmosphere will be taken from a United States standard mercurial barometer.

(9) Thermometers will be located as follows: No. 1 in barometer case to note the temperature of atmosphere. Nos. 2 and 3 in the two compartments of measuring-tank. No. 4 in calorimeter.

(10) The time of commencement and close of run and periods of observation will be taken from a chronometer clock placed near the pump under experiment; the periods of observation will be indicated by a double stroke of the signal-gong; one minute previous to each observation a single stroke of the gong will be made calling the assistants to their stations. Every (15) fifteen minutes a full set of observations will be made and entered in the log.

(11) A revolution counter will be connected to standard on piston-rod.

(12) Previous to experiments, each exhibitor will hand to the Board of Experts a complete schedule of dimensions of steam and water cylinder, internal diameter of steam and exhaust pipes, area of steam-ports, internal diameter of suction and delivery-pipes and volume of clearance in steam-cylinder.

(13) The economy will be determined by the water of condensation collected in the tank under condenser, corrected by the average of result obtained from calorimeter observations; and the cost of the work in coal (Pittsburgh No. 1) at $\frac{1}{8}$ this upon assumed boiler efficiency of (9) nine pounds water evaporated per pound of coal burned on the grates.

(14) The duty will be stated in gallons lifted one foot high.

90. Standard Systems of Boiler-trial have been already discussed and described at such length that it is unnecessary to add anything more here than to remark that, in every case of real importance, the careful and skilful management of the boiler-trial as a part of the whole work, in the measurement of the efficiency of the system of heat-production and utilization, becomes an essential element of success. The best standard

methods, as a whole and in every detail, should be adopted. Unless the measurement of the quantity and quality of the steam supplied be accurately made, it is quite impossible to obtain a correct measure of the efficiency of the apparatus in which it is utilized by conversion into work and power.

91. The Heat-energy, the Quantity and Quality of Steam used, and the availability of the heat stored in that steam and transferred to the engine with it, can only be exactly known when the weight, the wetness or dryness, the pressure, and the thermal properties of the steam are precisely ascertained. The weight of feed-water pumped into the boiler is the weight of the mixture of steam and water, if any water is entrained by the steam, which is sent to the engine. The heat so supplied, diminished by the usually simple waste from the exterior of the steam-pipe, is the amount received by that machine. The availability of that heat for its purpose depends upon the degree in which the temperature and the pressure of the steam exceed the temperature and pressure of the atmosphere or of the condenser.

The first point to be attended to is the testing of the steam to ascertain whether it be wet, dry and saturated, or superheated; and if not dry and saturated, to what extent it stores an excess per unit of weight by superheating, or a deficiency of heat in consequence of its admixture with water. This is determined by the use of the calorimeter.

Smoke-preventing apparatus is sometimes attached to boilers, and it becomes important to determine the quality of the products of combustion in this respect. This usually involves a boiler-trial and a comparison with ordinary furnaces. It will often be found that the prevention of smoke involves an excessive air-supply and a consequent waste of fuel and loss of efficiency. This may be partly compensated, however, by the improved performance of the boiler, due to cleaner and more effective heating-surfaces and the absence of soot deposits.

92. Special Methods of Engine-test are sometimes adopted in competitive trials of special forms, as illustrated by the following regulations prepared by Messrs. Hill and Holmes.

Code of Regulations for Tests of Automatic Cut-off Engines.

(1) Steam will be obtained from a pair of locomotive fire-box boilers, furnished by the Commissioners. These boilers have a combined evaporative capacity of 2250 pounds of water per hour; a heating-surface (combined) of 983 square feet; and a grate-surface of 12.7 square feet. Each boiler will be provided with a safety-valve, loaded to blow off at 85 pounds pressure above the atmosphere. Each boiler will have attached, independently, an accurate test-gauge, and if it can be procured, an Edson recording-gauge. The height of the water will be indicated by a glass water-gauge on each boiler, in addition to the usual test-cocks.

(2) The feed-water will be weighed in the receiving-tank, and drawn off as occasion requires into the supplemental tank. The water will be supplied to the boilers by an independent steam-pump, having the suction connected with the supplemental tank, and the discharge with check-valves of boilers. The steam to drive the boiler-feeders will be obtained from boilers independent of those furnishing steam for the engine under experiment. The water fed into the boilers will be determined in weight whilst in the receiving-tank. The receiving-tank will have a capacity of 2300 pounds water, at 175° Fahr.; the supplemental tank will have a capacity of 1000 pounds water at 150° Fahr.

(3) The resistance will be obtained by a 100 horse-power blower, having a sliding iron gate fitted to its discharge orifice. The position of the gate having been determined, it will be fastened at this point during the experiment.

(4) A pair of indicators will be attached to the cylinder of engine, one at each end. The indicators will be moved in such a manner that the diagrams shall be coincident with the motion of the piston.

(5) Two engine-counters will be employed, one to indicate the revolutions of the main shaft of engine, and one to determine the revolutions of the jack-shaft.

(6) The ... shaft, between the ... and the ...

(7) A steam-gauge ... near the ...

(8) The pressure ... ical thermometer ... the steam-pipe ... into the steam-...

(9) The temperature ... mercurial thermometer ...

(10) The temperature ... on a thermometer ... ering of the cylinder.

(11) The pressure ... U. S. standard ... phere will be ...

(12) The time ... placed near the engine ...

(13) The time ... a double stroke of the engine ... observation a single stroke ... assistants to their stations.

(14) Previous to experiments ... boilers will be carefully ... pipe connecting with engine ... pumps.

(15) Each exhibitor will hand to the Board ... previous to the experiments, a complete ... dimensions of his engine, including the ... steam and exhaust port area, and weight of ... parts.

(16) The duration of experiments will be ... hours. Previous to the beginning of experiments ... will be steamed up to the running pressure, and ... water brought to the thread tied around the ... tubes in the water-gauges. All water supplied to ...

thereafter will be weighed and charged to the engine. The height of water at close of experiment will be made to coincide with the thread on the glass tubes.

(17) Every fifteen minutes a full set of observations will be made and entered in the log.

(18) During the economy test, the engine will be run with full opening of stop-valve.

(19) The economy of the engine will be determined upon the consumption of water per I. H. P. per hour, and the cost of the power (in coal) one-ninth this upon an assumed evaporative efficiency of boilers of nine pounds of water per pound of coal.

Code of Regulations for Tests of Slide-valve Engines.

(1) Steam will be supplied from the boilers used in the test of automatic engines. The general dimensions are stated in paragraph 1 of the Code of Regulations for Experiments upon Cut-off Engines.

(2) The economy will be determined upon the consumption of water per I. H. P. per hour. The water will be delivered from the exhaust-heater into the receiving-tank, where it will be weighed and entered in the log of the engine under experiment. The water will be drawn from the receiving-tank into the supplemental tank connected with the suction of the pumps feeding the boilers. The steam to drive the boiler-feeders will be obtained from boilers independent of those furnishing steam for the engine under experiment. The receiving and supplemental tanks will be the same as used in the cut-off experiments. The dimensions are enumerated in paragraph 2 of the Regulations for the Test of Cut-off Engines.

(3) A calorimeter-test of the quality of steam furnished will be made regularly every 30 minutes.

(4) The power will be absorbed by a 100 H. P. pressure blower. This will have an adjustable gate fitted to the discharge-orifice to regulate the resistance. The area of opening will be fixed during the run.

(5) Diagrams from each end of cylinder will be taken. The

motion of indicator-drum will be such as to produce a diagram coincident with the movement of piston.

(6) The counter indicating the revolutions of engine will be connected direct. The counter showing revolutions of jack-shaft (carrying dynamometer) will be driven by positive connectors at a reduced speed.

(7) The dynamometer (transmitting) will be keyed to jack-shaft between the pulley receiving the belt from the engine, and the pulley carrying the blower-belt. The indications of the dynamometer will be read from a station in close proximity to the instrument.

(8) A test-gauge of approved make will be screwed into the steam-pipe as near the stop-valve as convenient. The initial pressure in the cylinder will be compared with the pressure in the pipe.

(9) Thermometers will be used as follows: No. 1 to show the temperature of atmosphere; 2, in the feed-water tank; 3, in the steam-chest of cylinder; 4, in the calorimeter. The thermometers to be U. S. standard instruments, thoroughly tested before the experiments, and of uniform scale. No. 1 will indicate temperatures from 32 to 120 degrees Fahr.; Nos. 2 and 4, from 32 to 250 degrees; and No. 3, from 32 to 600 degrees Fahr.

(10) The pressure of atmosphere will be read from a U. S. standard compensated aneroid barometer.

(11) The time of commencement and close of run, and intervals of observations for the log will be taken from a chronometer clock, placed near the engine under experiment. The time of noting observations will be indicated by a double stroke of the signal-gong. One minute previous to each observation a single stroke of the gong will be made calling the assistants to their stations.

(12) Previous to the experiments all pipe connections with boilers will be carefully closed, leaving open only the steam-pipe connecting with the engine and feed-pipe connecting with the pumps. Great care will be taken that all steam generated in the boilers be delivered to the engine.

(13) The duration of run will be fixed at five (5) hours. Previous to the beginning of experiment, the boilers will be steamed up to the running-pressure (75 lbs. above atmosphere), and the height of water brought to the thread around the tube in glass water-gauge. All water delivered to the boilers thereafter will be regularly entered, by weight, in the log. The condition of pressure and height of water will be maintained as nearly uniform as possible during the run, and made to coincide with the initial conditions at close of run.

(14) During the economy-test the engine will be operated with an open stop-valve.

(15) At close of economy-run the main belt will be thrown off, and the engine throttled to run at load speed for the friction-diagrams.

(16) Previous to the experiments each exhibitor will hand to the Board of Experts a complete summary of the dimensions of his engine, including volume of clearance, steam, and exhaust-port area (least), weight of reciprocating parts, and estimated I. H. P., at 75 pounds pressure in the pipe.

(17) Every fifteen minutes a full set of observations will be made and entered in the log.

(18) The cost of the power in coal will be taken at one ninth the consumption of water per I. H. P. per hour upon an assumed boiler efficiency of nine pounds water evaporated per pound of coal burned on the grate.

Code of Regulations for Tests of Mounted (Portable) Engines.

(1) Steam will be furnished each engine by its attached boiler. Each exhibitor will be required to hand to the Board of Experts, previous to the experiments, a schedule of the length, width, and height from grates to crown of fire-box; total heating-surface; number, length, and external diameter of tubes; thickness of water-leg; diameter and vertical and horizontal distance apart of stay-bolts, diameter and length of barrel, total area of openings through fire-bars, area of openings through ash-pit door, thickness of iron in shell and fire-box,

cubic feet of water carried to the gauge-line, cubic feet of steam room.

(2) Before the commencement of experiments the boiler will be steamed up to the running pressure, using ......for fuel. Each exhibitor will have weighed to him a sufficient quantity for the economy-run. The management of the fire and the use of the fuel will be entirely under the control of the exhibitor. All fuel remaining in the pile at close of run, and all unburnt wood on the grates, will be weighed back to the credit of the exhibitor. The ashes under the grate will be weighed back dry.

(3) A calorimeter-test of the quality of steam furnished by the boiler will be made regularly every 30 minutes.

(4) The dynamometer will be keyed to main shaft of engine; the power will be taken off from a pulley attached to loose side of dynamometer; the resistance will be created by a 40 H. P. pressure blower, the discharge-orifice of which will be fitted with an adjustable gate; the area of discharge-opening will be fixed constant during the run.

(5) One revolution-counter will be employed, having a positive connection with the valve-stem of engine.

(6) The feed-water will be weighed in the receiving-tank, and passed thence to the supplemental tank, from which it will be drawn by the boiler-feeder. The water will be delivered from the city mains (unheated) into the receiving-tank. Each engine will heat its own feed-water.

(7) The duration of run will be fixed at five (5) hours. Previous to the experiment, the engine will be run without load until all parts have acquired the working-temperature and the water brought to the thread tied around the tube of glass water-gauge. All water fed to the boiler from commencement to close of run will be regularly weighed and entered in the log. The pressure of steam and height of water at close of experiment will be made to coincide with the initial conditions.

(8) Thermometers will be located as follows: No. 1, in

barometer-case, to indicate the temperature of atmosphere; No. 2, in feed-water tank; No. 3, in calorimeter.

(9) The pressure of the atmosphere will be taken from a U. S. standard compensated aneroid barometer.

(10) The pressure of steam in the boiler will be read from a reliable steam-gauge, independent of that belonging to the engine.

(11) Previous to experiment, all pipe connections with boiler or engine will be detached, except the pipe from steam-dome to cylinder, and the suction-pipe from feed apparatus to supplemental tank.

(13) The economy will be determined upon the consumption of coal per I. H. P. per hour.

(14) Before the experiments begin, each exhibitor will hand to the Board of Experts a complete summary of the dimensions of engine, including volume of clearance, area of steam and exhaust ports, internal cross-section of steam-pipe, and weight of reciprocating parts. The valve-motion will be shown by a single indicator-diagram from each end of the cylinder, taken at a uniform piston-speed.

(15) During the economy-run the engine will be operated with an open stop-valve.

(16) At close of economy-run the main belt will be thrown off, and the engine throttled to run at load speed. Friction-diagrams will then be taken from each end of cylinder.

(17) Every fifteen minutes a full set of observations will be made and entered in the log.

(18) The time of commencement and close of run, and period of observation, will be determined by a chronometer clock placed near the engine under experiment. The time of noting observation will be indicated by a double stroke of the gong. One minute previous to each observation a single stroke of the gong will be made, calling the assistants to their stations.

CHAPTER IX.

EXAMPLES OF ENGINE-TRIALS.

93. **Examples of Engine-testing,** as illustrating current and standard practice, will better complete a treatise on this subject than any further extended descriptions of details and of methods of observation, of computation, and of preparation of reports. In the following pages are given, as fully as is possible without occupying too great space, illustrations of this character, as obtained by reference to the reports of the more expert and most experienced of contemporary engineers, or to reports of earlier work which have been regarded by engineers as best representing good practice in special departments.

In all cases the engineer must himself judge whether to err, if at all, in making such tests and in preparing his report, in the direction of extended and complete—and hence costly—investigation and deduction, or in that of brevity and possible incompleteness. The general rule should be to secure all data essential to the purposes of the trial, and, incidentally, to secure all additional facts and data that he may find attainable without incurring objectionable expense. If, in any case, a doubt arises, it will usually be wisest to err on the side of completeness and accuracy. Thus, when called upon, as an expert, to ascertain whether the terms of a contract specifying simply the duty of a pumping-engine have been fully complied with, he need only measure, usually, the coal consumption, the quantity of water pumped by the machine, and the head against which the water is raised. To attempt more would sometimes be unnecessary, and might involve considerable unauthorized expense. But the expert engineer may often, without appreciable additional cost, obtain valuable data relating to the distribution, the utilization, and the wastes of heat, of

energy, and of steam and fuel: he should in such case endeavor to obtain all such data and make them useful.

The following selected illustrations cover the ground very completely, and will probably be quite sufficient for their purpose.

94. Illustrations of the Trial of Stationary Engines abound in the current periodicals, and may be referred to by the engineer seeking to make comparisons. As a model of brevity in reporting such a trial the following is selected. It is the report of Mr. Flower on a trial of a small Corliss engine designed by Mr. Edwin Reynolds.

"GENTLEMEN: I have made an economy test of your engine and boilers as directed by you, and beg leave to report thereon as follows:

Date.	Test of engine made	Aug. 3d, 1882.	
Description of Engine.	Reynolds' Corliss		
	Diameter of cylinder	14 in.	
	Length of stroke	36 "	
	Clearance (assumed)	.025	
	Nominal horse-power at 82 Rev	68	
Trial.	Trial began at	7.30 A.M.	
	" ended "	4.06 P.M.	
	Duration of trial	8 h. 36 m.	
Revolutions.	Revolutions of engine during test	35,372	
	" " per minute	69.22	
	Temperature of engine-room	99°	
Gauge-pressure.	Maximum pressure in boiler	72	pounds.
	Minimum " "	52	"
	Average " "	65	"
	Variable boiler-pressure	80	"
Pressure per Indicator.	Average initial pressure on piston	61.2	"
	" total " "	27.08	"
	" gross effective pressure on piston	25.58	"
	Per cent. of boiler-pressure appearing as initial	94	
Counter-pressure per Indicator.	Back-pressure due to friction through ports	.45	pounds.
	" " " cushion	1.05	"
	" " total on piston	1.50	"
Indicated Horse-power.	Maximum gross effective horse-power	66.00	
	Minimum " " " "	41.40	
	Average " " " "	49.30	
	Average total" " " "	52.20	

Friction Diagrams.	Maximum friction horse-power............	27.21
	Minimum " " "	22.88
	Average " " "	24.32
Distribution of Load.	Average total horse-power................	52.20
	" gross effective horse-power.......	49.30
	" net " " "	24.98
Dynamo, Horse-power of.	Edison's dynamo required................	4.58 h. p.
	Brush " "	11.54 "
Water used, actual.	Temperature of feed-water................	203°
	Total water pumped into boiler............	11,520.5 pounds.
	" " " " " per hour....	1,340.0 "
	Moisture in steam........................	3 per cent.
Steam per hour, actual.	Dry saturated steam per hour..............	1300 pounds.
	Dry steam per indicated horse-power, per hour............................	24.9 "
	Dry steam per gross effective horse-power per hour..........................	26.36 "
Steam used, as shown by Indicator.	Steam accounted for by indicator per hour per horse-power, total................	24.37 "
	Efficiency of cylinder.....................	.97
Coal per hour per H. P.	Coal per hour per horse-power, anthracite...	3.03
	Combustible " " "	2.60
Cost.	Cost per day of 10 hours, coal @ $6.25 ton.	$4.94
Coal.	Coal " " anthracite.......	158 pounds.
Indiana block coal.	Coal per hour per horse-power............	3.62 "
	" per day of 10 hours..................	1,890 "
	Cost " " @ $3.75 ton......	$3.54
Maximum Diagram No. 31.	Indicated power of heaviest diagram......	66 h. p.
	Cut-off in parts of stroke..................	3.60
	" from commencement of stroke.....	10 inches.
	Initial pressure..........................	64 pounds.
Possibilities of Engine.	Same initial pressure as above............	
	Diagram, ¼ cut-off, engine will develop....	72 h. p.
	Same cut-off as No. 31, and 74 lb. initial pressure..........................	70 h. p.
	Same cut-off and same initial pressure as No. 31, with 82 rev. per minute will develop..	77 "

"When No. 31 was taken, all the machinery that could be put on was on. The average power used was much less than No. 31, being 49.3 horse-power.

... ...port, the power of the engine ... the revolutions to 82 per ... speed as given by the builder), ... by a higher initial pressure.

... may be had by increasing the ... during test was 65 pounds, and ... by the inspector.

...ure and a more uniform pressure than ... rable.

... onomy of an engine is the amount of ... er horse-power. From that point of ... above the average.

... ngine increases above 53 to 55 horse-... will decrease, unless the initial pressure ... speed of revolution is given to engine.

... engine by coal is not as good as it should ... proved; 3.03 lbs. coal (anthracite) per hour ... high, but the fault is not with the engine, ... per hour per horse-power is only 24.9, and ... lbs. actual would give an indicated horse-... coal; the boiler should evaporate 10 lbs. ... thracite coal."

... is an example of a very full report by Mr. ... of a Corliss engine designed and built by Mr.

... gine, 24″ diameter of cylinder and 60″ stroke of ... ondensing, and fitted with the ordinary jet-con-... reciprocating air-pump. The injection-water is ... lift of 15′ from the Mississippi River, upon the ... the mill stands; and during the trial the con-... entered the injection-pipe at a temperature near ... point. The steam-valves were formerly closed by ... weights; but previous to the trial, vacuum dash-pots ... to insure a prompt closing of the valve when liber-... the hook. The engine is furnished with a pulley ... diameter and 32″ face; driving back to the line-... with a ...′ double leather belt.

"The exhaust of engine is closely connected to a condenser by a 10″ pipe, and steam is conveyed from the boiler by a 7″ pipe.

"Steam is furnished by a pair of tubular boilers set in battery, and each of the following dimensions: 60″ diameter of shell, 12′ long fifty 4″ tubes. Each boiler is fitted with a vertical steam-dome, 30″ diameter × 36″ high, and over these and joined to them by short legs is a horizontal steam-drum, 24″ diameter and 14″ long.

"The steam-pipe is joined by branch pipes to the side of the horizontal drum.

"The feed-water is taken from a drop-leg in the overflow-pipe from the condenser, and conducted to the suction of a single-acting plunger-pump driven from the engine by belt. Into the breeching or front smoke connection has been introduced a fuel economizer, consisting of 250′ of 2¼″ iron pipe, through which the feed-water is forced to the boiler.

"The furnace is arranged to burn slabs and hard wood, although by the record it would appear to be well adapted for coal (the fuel used during the trial of engine). The lack of a suitable bridge wall, and the very large furnace-doors and grate-surface are not calculated for maximum economy with coal as a fuel, and it is eminently probable that with a different construction of furnace the efficiency of the boilers during the trial of engine would have been higher.

"The entire net power of engine is expended in driving the machinery of the mill, which consists of twelve run of 54″ buhrs, and three run of 48″ buhrs; two crushing rolls, each with 3-12″ × 30″ cylinders; five rolls, each with 2–12″ × 30″ cylinders, and one roll with 2–12″ × 18″ cylinders. The bolting machinery consists of one chest with two reels; two chests with three reels; one chest with six reels, and one chest with eight reels; in all twenty-two bolting reels and forty-eight conveyors.

"The cleaning machinery consists of two 'cockle' machines; one 'scouring' machine; one 'separator,' and two brushing machines. Of the purifying machines there are seventeen;

and one shaking machine; four flour-packers; four stand of wheat elevators; four stand of flour elevators, and twenty-one middlings elevators; one small and two large exhaust fans.

"To this should be added the machinery of the grain elevator, which is driven by belt from the third story of the mill; and the line-shafting, connecting belts, pulleys, and gearing, forming the general machinery of the mill.

"In the following tables are given the principal measured and calculated data of engines and boilers. The clearance was not measured, but estimated at three per cent. of piston-displacement, this being the usual clearance in similar engines of like dimensions.

"The factor of horse-power due mean area, and velocity of piston for each mean effective pound pressure, has been calculated as follows: The area of a 24" piston is 452.39 sup. ins. and the area of the rod (3.375") is 8.9462 sup. ins.; and the mean area of piston is, therefore,

$$452.39 - \frac{8.9462}{2} = 447.917 \text{ sup. ins.},$$

and the factor of horse-power

$$\frac{447.917 \times 596.166}{33000} = 8.20446.$$

"The valve functions have been measured on the diagrams. The volume of steam accounted for to release is obtained by taking the mean area (feet) of piston into the piston travel (feet) per hour to point of release, to which is added the hourly volume of clearance. The volume of steam retained by exhaust-closure is obtained by taking the mean area of piston, in feet, into the travel of piston, in feet per hour, from exhaust-closure to end of stroke, to which is added the hourly volume of clearance.

"The dimensions of boilers and fire-grates are furnished by your engineer, from which have been deduced the heating-surface, grate surface, and calorimeter of tubes, and ratios of

heating to grate-surface, and grate-surface to cross-section of tubes.

Dimensions of Engine.

Diameter of cylinder....................	24 inches.
Stroke of piston..........................	60 "
Revolutions per minute during trial.......	59.616.
Piston speed " " " "	596.166 feet.
Factor of H. P. due area and velocity of piston	8.204.
Piston stroke to release in parts of stroke..	99.370.
" " exhaust closure in parts of stroke	6.067.
Clearance (estimated) in parts of stroke.....	3.000
Volume of steam to release per hour......	115,038.04 cu. ft.
" " retained by cushion per hour........	10,189.02 cu. ft.
Diam. of air-pump........................	12 inches.
Stroke " "	15 "
Diameter of driving-pulley...............	20 feet.
Face " " "	32 inches.
Weight " " "	40,000 pounds.

Dimensions of Boilers.

Number.................................		2.
Diameter of shells.......................		60 inches.
Length " "		12 feet.
Tubes, each boiler.......................		50–4 inches.
Heating-surface shells (2).......	250.56	
" " tubes (100).....	1,245.64	
" " heads (4)......	40.72	
" " total....................		1,536.92 sup. ft.
Grate "		51.75 sup. ft.
Calorimeter of flues		1,256.64 sup. ins.
Heating to grate-surface..................		29.70
Grate-surface to calorimeter..............		5.93

"The trial of engine for economy of performance and trial of boilers for evaporative efficiency were made simultaneously; all preparations having been completed, the trial began at 9.15 a.m., and terminated at 7.15 p.m.; duration of trial, ten hours.

"The test of boiler efficiency was with coal.

"The load was that usually carried in the daily operation of the mill, and through the care of the chief miller, was held quite uniform during the ten-hours' run. It is possible that the mean power developed is slightly greater than usual, from the fact that the operatives were cautioned to avoid breaks in the load; and that they obeyed the injunction is best attested by the indicator-diagrams, which exhibit but slight variations in the load during the economy-trial.

"The diagrams were taken by independent indicators, one to each end of cylinder. Forty (40) pound springs were used, and the drums were moved by well-constructed bell-cranks and reciprocating connections hung on a stout gallows frame. The joints of the levers and connections were carefully made, and means were provided to take up wear and avoid lost motion.

"The strings on the indicator-barrels were only long enough to couple with the pins on the short stroke-reciprocating-bar, and the recoil springs were adjusted as nearly as possible to the same tension. The length of diagrams was uniformly 4''.78.

"During the trial a pair of diagrams were taken regularly every fifteen minutes, making eighty-two diagrams; from which have been obtained the initial pressure in cylinder, piston-stroke to cut-off, ratios of expansion by pressure and by volumes, terminal pressure, counter pressure at mid-stroke, utilization of vacuum, and mean effective pressure on the piston, from which is obtained the mean power developed.

"The vacuum in the condenser and the pressure in the boilers were taken from gauges in the engine-room regularly every fifteen minutes.

"The temperature of water to the condenser was taken in the river at the mouth of the injection-pipe. The temperature of overflow from the condenser was taken in the measuring-tank. The temperature of feed to the boiler was taken in the feed-pipe near the check-valves.

"The water delivered to the boilers was measured in the following manner: Two oil-barrels were carefully washed inside and placed on the same level in the engine-room; to the bottoms of these was connected, by branch pipes, the suction-pipe of pump, each branch being provided with an open way-cock to shut off the flow when the level had been reduced to the lowest gauge-point. The pipe from the hot-well to the pump was cut and carried out over the barrels; a connection made by branches to each barrel, and a stop-valve in each branch regulated the flow of water into the tanks. The tanks or barrels were numbered 1 and 2, and were alternately filled to the overflow notch in the rim, and emptied to the centre of the branch pipe in the side of barrel, and the contents discharged into the pipe leading to the pump.

"Whilst the No. 1 barrel was running out, the No. 2 barrel was filling with water from the hot-well; and directly the first barrel was emptied to the lower gauge-point, it was turned off and the second barrel turned on; and so on during the entire trial, the empty barrel being shut off before the full one was turned on, to prevent transfer of water from the full to the empty barrel. Directly each barrel of water was turned on, the time was entered in the log, and a tally made by the assistant in charge of the tanks. From time to time my record of tanks discharged was compared with the assistant's tally, to avoid error in the count.

"After the trial, the capacity of each tank was determined by filling to the overflow notch, noting temperature, drawing off to the lower gauge-point, and weighing. The temperatures of the tanks of water discharged into the suction-pipe of feed-pump having been regularly noted during the trial, the weight of water delivered to the boiler was deduced from the number

"Th... tanks at mean observed of boi... all ... 9.15 ... hou...

... water entrained were made by ... near the pipe to the engine, ... condensing it in a given ... perature of the water before ... n, and the pressure of evapo- th... was made. The thermal val- ... ure and the weights of steam ... thermal values of saturated steam ... tuted the data from which has ... resident in a pound of evapora- ... which has been deduced the water ... 2.84 per cent. of the total water ... twenty calorimeter-observations were ... trial.

... the engine are nominally 60 per min- ... hours' continuous record by counter, the ... minute were 59.616.

... during trial of engine was Wilmington, ... part of Illinois, and, from the evapora- ... oped, of very fair quality.

... fires were cleaned before the trial, and ... accumulated during the ten hours' firing ... The non-combustible by weight constituted ... the total coal fired. Previous to commence- ... water-level in both boilers was marked on the ... the fires levelled and thickness noted; the ... of fires and water-level obtained at the end of

... following tables are given the observed and calcu- ... the performance of engine and boilers. ... the diagrams are means of eighty-two readings, ... are means of forty-one readings.

... of engine by steam and by coal is developed ... quantities charged per hour.

Data from Trial of Engine.

Duration of trial	10 hours
Mean pressure by boiler-gauge above atm	92.876 lbs.
" initial pressure above atm	89.376 "
" terminal " absolute	12.018 "
" counter " "	2.696 "
" cut-off in parts of stroke apparent	15.560
" " " " actual	18.019
" vacuum by gauge	26.40 inches
" " " diagrams	24.05 "
" temperature of injection	33.840
" " of hot-well	92.725
" effective pressure	32.9792 lbs.
Indicated horse-power	270.5796
Ratio of expansion by volumes	5.549
" " " " pressures	8.643

Economy of Engine.

Total water per hour to boilers	5,037.128 lbs.
Water (steam) per hour to calorimeter	10.000 lbs.
" entrained per hour in the steam	655.583 lbs.
Net steam per hour to engine	4,371.545 lbs.
Steam per indicated horse-power, actual,	16.156 lbs.
" " " " by the diagrams	13.035 "
Percentage of steam accounted for	80.682
Coal burned per hour	535. lbs.
Coal per indicated horse-power per hour	1.9772 lbs.
" " " " evaporation 9 to 1	1.7950 "
Combustible, per indicated horse-power, per hour	1.8328 "

Performance of Boilers.

Duration of trial	10 hours
Pressure by gauge	92.876 lbs.
Temperature of feed to heater	92.725
" " " " boiler	114.324
Elevation of feed by heater	21.599
Percentage of gain by heater	1.723

...boilers	50,371.28	lbs.
...steam (12.84 %)	6,467.70	"
...	43,903.58	"
...	5,350	"
...weighed back	390	"
...	4,960	"
...	8.206	"
...combustible	8.852	"
...from temp. of 212° and pres. of atm.	9.639	"
...foot of heating-surface	.022	"
... " grate-surface	10.300	"
...in coal	7.3	

...during trial, Wilmington (Illinois).

...economy-trial of engine, the flour manufactured ...report, 217 barrels high grade, and 2 per ...low grade, or 221.34 barrels produced in ten ...indicated power of engine was 270.56 horse-...the hourly expenditure of power per barrel of flour

$$\frac{270.56}{22.134} = 12.224 \text{ H. P.}$$

...coal burned for whole trial was 5350 pounds, and coal ...flour produced becomes $\frac{5350}{221.34} = 24.198$ pounds.

...the experiments (firing, slabs and hard wood) were ...the engine was indicated for distribution of the ...the mill.

...first (*A*) load was with all the machinery on, and op-...under the ordinary conditions. The second (*B*) load ...all the machinery on except the machinery in eleva-...ting. The third (*C*) load was with all the machinery ...except the flour-packers. The fourth (*D*) load was with ...machinery on except the cleaning machinery and flour-packers. The fifth (*E*) load was with all the machinery on ...the grinding-rolls. The sixth (*F*) load was with all the machinery on except the purifiers. And the seventh (*G*) load was with all the machinery on except the grinding-buhrs.

"The changes of load were made quickly, in order to preserve the conditions of ordinary performance in the special machinery driven; and the power developed for each load has been estimated from six diagrams, three from each end of cylinder.

"The indicated loads were as follows:

First load (*A*)	267.503	H. P.
Second load (*B*)	262.585	"
Third load (*C*)	363.706	"
Fourth load (*D*)	250.726	"
Fifth load (*E*)	246.740	"
Sixth load (*F*)	242.645	"
Seventh load (*G*)	117.149	"

"Each of these loads is made up of the friction of engine in all parts; extra friction of engine due to the load, friction of all the driving machinery in the mill, and power required to drive the special machinery, including friction; in like manner the differences between the maximum load and reduced loads nearly represent the power required to drive the special machinery not on, including its own friction.

"Hence the difference between the maximum load and lesser loads represents slightly more than the power actually absorbed by the special machinery not driven.

"The distribution of the power in the mill is thus as follows:

Total indicated power of engine load (*A*)		267.503
Friction of engine alone	16.409	
Extra friction due load	12.544	
Grinding-buhrs	150.354	
Cleaning machinery	12.980	
Elevator	4.918	
Crushing-rolls	20.763	
Bolting-reels, conveyors, fans, and general machinery	21.860	
Middlings-purifiers	23.868	
Flour-packers	3.797	
		267.503

"I have attached to this report one pair of diagrams taken during the trial, and numbered 14, upon which have been drawn theoretic curves from the terminal pressures and points of release. The lack of coincidence between the actual and theoretic curves is due, in my opinion, to a slight leak past the piston or out of the exhaust; in all probability, the latter. The engine being very new, and a certain amount of wear being requisite to make good joints between the valve faces and seats, it is probable that this leak will in time remedy itself. However, it does not appear that the failure of the diagram to satisfy the conditions of theory has any marked effect on the economy of the engine, for the actual consumption of steam and coal per indicated horse-power per hour are the least I have ever obtained from a single-cylinder engine."

The preceding illustrations exhibit the methods of report and the results of trial of good examples of ordinary practice with the common simple engine of moderate size and under usual working conditions.

As illustrating what can be done with a good compound stationary engine, the following data are presented, as given the Author by Mr. Corliss a short time before his death, as the results obtained from a compound condensing-engine of about 500 I. H. P., driving a cotton-mill:

RESULT OF A SEVEN DAYS' TRIAL OF THE CORLISS COMPOUND ENGINE AT NOURSE MILL.

Commencing Oct. 15th, and ending Oct. 22d, 1884.

	Starting Fires.			Coal while running.	Wood used during running time.	Value of same at 40 %.	Total new fuel fed to furnace each day.	Cinders used during day.
	Wood.	Value at 40 % of Coal.	Coal.					
Wed., Oct. 15, 1884..	1000	400	2647	6858	200	80	9905	
Thur., " 16, " ..	833	333	2535	7200			10068	
Fri., " 17, " ..	30	12	2800	7200			10012	
Sat., " 18, " ..			3200	3000			6800	1671
Mon., " 20, " ..	800	320	3200	7200			10720	
Tue., " 21, " ..	10	4	3200	6004			10208	
Wed., " 22, " ..			3200	4300	30	12	7512	3018
Totals and averages for 7 days........	2673	1069	20782	43242	230	92	65135	

	Total fuel fed to furnace each day.	Cinders found at end of day	Total Fuel used.	Running Time.	Fuel used per hour.	Indicated Horse-power.	Fuel per hour per I. H. P.	Average Rev.
Wed. Oct. 15 1884...	9965	863	9102	11 H. 44 M.	775.76	498.93	1.55	56.96
Thur. " 16 " ...	10068	685	9383	11 " 41"	803.13	496.37	1.61	57.10
Fri. " 17 " ...	10012	730	9282	11 " 40"	795.64	495.09	1.60	57.08
Sat. " 18 " ..	8471	729	7742	9 " 44"	795.41	483.80	1.64	57.46
Mon. " 20 " ...	10720	742	9978	11 " 42"	852.82	509.46	1.67	56.84
Tue. " 21 " ...	10108	717	9391	11 " 40"	804.98	504.24	1.59	56.99
Wed. " 22 " ...	10530	455	10075	11 " 41"	862.36	501.94	1.71	57.17
Totals and averages for 7 Days........				79 H. 52 M.	812.38	499.13	1.62	57.10

General Summary of 7 Days' Trial of Nourse Mill Engine.

Commencing Oct. 15th, and ending Oct. 22d, 1884.

Fires were started on clean grates Wed., Oct. 15th, 1884.	
Wood 2903 lbs. Equivalent value in Coal at 40 per cent., ...	1161 lbs.
Coal for starting fires, ...	20782 "
Coal used while running, ...	43242 "
Total amount of fuel fed to furnaces, ...	65185 "
Deduct cinders found at end of run, 455 lbs.; estimated at 80 per cent. value of new coal, .	364 "
Total fuel used during the seven days, ...	64821 "
Running time, ...	79 H. 52 M.
Fuel per hour, ...	811.62 lbs.
Average indicated horse-power from 636 cards, .	499.13
Fuel per hour per indicated H. P., ...	1.626 lbs.
Average revolutions per minute, ...	57.1
Per cent. of ash and clinker, ...	11.9

The results of a series of trials of a single-acting compound engine, as given in the following table, illustrate well the system and the conciseness which characterize the work of the expert engineer, as well as the efficiency attainable with this class of engines when well designed, well constructed, and operated under favorable conditions :

STEAM PER INDICATED H. P. SINGLE-ACTING COMPOUND ENGINE.

Non-condensing.					Condensing			
Boiler-pressure.				Horse-power.	Boiler-pressure.			
60 lbs.	80 lbs.	100 lbs.	120 lbs.		120 lbs.	100 lbs.	80 lbs.	60 lbs.
			22.6	210	18.4			
		23.0	21.9	170	18.1	18.8		
	24.9	23.6	22.2	140	18.2	18.5	20.0	
	25.7	23.9	22.2	115	18.2	18.6	19.6	20.5
26.9	25.2	24.9	22.4	100	18.3	18.6	19.7	20.3
27.7	25.2	25.1	24.6	80	18.3	18.6	19.9	20.1
30.3	28.7	29.4	28.8	50	20.4	20.8	20.7	20.4

The engine was of 14 and 24 inches diameter of cylinder, 14 inches stroke of piston, and unjacketed. All steam passing into and through the engine was weighed and measured. Gauge-pressures are given. This engine is usually rated at 150 H. P. The steam was probably dry, but not superheated.

95. **Tabulated Deductions and General Conclusions** from engine-trials, as above illustrated, should usually be presented in as concise and legible form as possible, and so arranged as to make it easy to interpret the data and to verify the results both as to facts and reasoning. Thus, one of the most complete investigations ever made in this field was that of Mr. Willans on the efficiency and wastes of his triple-expansion engine.* He describes in his report a series of economy-trials, non-condensing, made with one of his central-valve engines, with one crank, having three cylinders in line. By removing one or both of the upper pistons, the engine could be easily changed into a compound or into a simple engine at pleasure.

Check-trials were made by Mr. MacFarlane Gray, Prof. Kennedy, Mr. Druitt Halpin, Professor Unwin, and Mr. Wilson Hartnell. The work theoretically due from a given quantity of steam at given pressure, exhausting into the atmosphere, was first considered.

By a formula of Mr. MacFarlane Gray, which agreed with the less simple formulas of Rankine and Clausius, the weights of steam required theoretically per indicated horse-power were ascertained.

A description is given of the main series of trials, of the ap-

* Proc. Brit. Inst. C. E., Mch. 13, 1888. Sci. Am. Supp., May 20, 1888.

pliances used, and of the means taken to insure accuracy. The missing quantity of feed-water at cut-off, which, in the simple engine-trials, rose from 11.7 per cent. at 40 lbs. absolute pressure to nearly 30 per cent. at 110 lbs., and at 90 lbs. was 24.8 per cent., was at 90 lbs. only 5 per cent. in the compound trials. In the latter, at 160 lbs., it increased to 17 per cent., but on repeating the trial with triple expansion, it fell to 5.46 per cent. or to 4.43 per cent. in another trial not included in the table.

The compound engine must always give a smaller diagram, considered with reference to the steam present at cut-off, than a simple engine, and a triple a smaller diagram than a compound engine. But even at 80 lbs. absolute, the compound engine had an advantage, not only from reduced initial condensation, but from less loss from clearance, and from reduced leakage. These gains became more apparent with increasing wear. The greater surface in a compound engine had not the injurious effect sometimes attributed to it, and the author showed how much less the theoretical diagram was reduced by the two small areas taken out of it in a compound engine than by the single large area abstracted in a simple engine. The trials completely confirmed the view that the compound engine owed its superiority to reduced range of temperature. At the unavoidably low pressures of the trial, the losses due to the added passages, etc., almost neutralized the saving in initial condensation; but with increased pressure—say to 290 lbs. absolute there would be considerable economy. The figures of these trials showed that the loss of pressure due to passages was far greater with high- than with low-pressure steam, and that pipes and passages should be proportioned with reference to the weight of steam passing, and not for a particular velocity merely.

After comparing the data of initial condensation in cases where the density of steam, the area of exposed surface, and the range of temperature were all variables, with other cases (1) where the density was constant and (2) where the surface was constant, the author concluded that, at four hundred revolutions per minute, the amount of initial condensation depended chiefly on the range of temperature in the cylinder, and not

upon the density of the steam or upon the extent of surface, and that its cause was probably the alternate heating and cooling of a small body of water retained in the cylinder. The effect of water intentionally introduced into the air-cushion cylinder showed how small a quantity of water retained in the cylinder would account for the effects observed. At lower speeds, surface might have more influence. The favorable economical effect of high rotative speed was very apparent.

Starting with 40 indicated horse-power, 130 lbs. absolute pressure, four expansions, and a consumption of 20.75 lbs. of water, the plan of varying the expansion, as compared with throttling, showed a gain of about 7 per cent. at 30 indicated horse-power, but of a very small percentage when below half-power. If the engine had an ordinary slide-valve, the greater friction, added to irregular motion, would probably neutralize the saving; while if the engine were one in which initial condensation assumed more usual proportions, the gain would be probably on the side of variable pressure. The diagrams showed that the missing quantity became enormously large as the expansion increased. Judging only by the feed-water accounted for by the indicator, the automatic engine appeared greatly the more economical, but actual measurement of the feed-water disproved this. The position of the automatic engine was relatively more favorable when simple than when compound.

The tabulated figures are given on page 349.

M. Delafond, testing a single-cylinder Corliss engine, built at Creusot, with similar care and thoroughness, comes to the following conclusions as fairly deducible from the data so obtained:*

(1) The effective work is equal to

$$T_e = -\alpha + \beta T_i;$$

where T_i is the indicated work; but the coefficients α and β are not absolutely constant, varying with the pressure of steam.

* Essais Effectuées sur une Machine Corliss, etc.; Paris, Dunod, Editeur, 1884; p. 60.

TRIALS OF TRIPLE EXPANSION ENGINES.

Intended mean admission pressure ...Lb.	40	90		110		130	150		160		170
Simple, Compound or Triple.	S.	S.	C.	S.	C.	C.	C.	T.	C.	T.	T.
Actual mean admission pressure...Lb.	40.87	92.65	87.54	106.3	109.3	130.6	149.9	151.9	158.5	158.1	172.5
Percentage ratio of actual mean pressure, referred to low-pressure piston, to theoretical mean pressure............	98.26	100	91.3	100.7	94.8	94.2	94.6	84.54	95.9	85.3	85.2
Ind. horse-power.....	26.51	31.61	29.14	33.5	33	36.31	38.59	35.69	39.55	35.36	38.45
Feed-water actually used per indicated H. P. H.—											
Simple . . Lb.	42.76	26.89	..	26	....	...	....	....	....	...	...
Compound.. Lb.	...	...	24.16	...	21.37	20.35	19.45	...	19.19	...	..
Triple .. ..Lb.	..		....	..	...	..					
Steam required theoretically per I. H. P. H. ... . . Lb.	34.67	19.24	19.86	17.9	17.65	16.25	... 15.23	19.68 15.16	.. 14.87	19.19 14.0	18.45 14.36
Percentage efficiency.	81.2	71.5	82.2	68.8	80.5	80.0	78.3	77	77.4	77.6	77.8
Percentage of feed-water missing at cut-off in high-pressure cylinder .	....	....	....	....	...	...	.	5.33	.	6.84	5.01
Ditto intermediate cylinder.	....	...	5	....	9.5	11.7	15.1	14.84	17	12.06	15.33
Ditto low-pressure cylinder	11.7	24.8	25.2	29.56	16.25	19.1	20.6	22.12	21.3	22.11	24.21
Percentage of feed-water missing at end of stroke in low-pressure cylinder...	10.4	18.83	24.25	21.53	16.59	27.55	20.69	18.01	19.55	18.81	19.25

(2) The best economy was measured under the following conditions:

With condenser;
" jacket;
" moderate pressure;
" " expansion.

In the best cases, the expenditure of steam, per effective and per indicated horse-power, per hour, was respectively $9^k.50$ and $7^k.75$ (about 17 and 21 pounds).

(3) The steam-jacket is advantageous for high ratios of expansion and high pressures; its effect decreases with reduced pressure and expansions, and becomes insignificant at low pressures and small ratios of expansion.

(4) Compression is useful in non-condensing engines, and

the more so as the final pressure is made to approximate initial steam-pressure.

(5) The best results were obtained in these cases at 120 to 170 I. H. P. At higher powers the cost in steam rapidly rises. Above 175 horse-power, the condenser is of no advantage, and high compression and the use of a good feed-water heater are advisable.*

(6) Initial condensation increases with increase of pressure, and lessens with diminishing expansion, becoming insignificant at full-stroke. The jacket reduces this loss; but the presence or absence of the condenser has no effect, this cylinder condensation being a result of expansion.

(7) The cylinder condensations and re-evaporations are of complex character; the jacket increases the re evaporation; pressure of 3½ or 4½ atmospheres gave largest evaporations.

(8) Moderate pressures are best; † as they give small initial condensation and considerable re-evaporation.‡

(9) High piston-speed and the use of steam in the jacket of higher pressure than in the engine are advisable.

(10) Working non-condensing at pressures of 3½ to 5½ atmospheres, with small expansion, the permanent presence of water in the cylinder could not be detected.

The following tables, prepared by Mr. Isherwood from the data obtained by a committee of the Société Industrielle de Mulhouse at a series of trials of a condensing compound engine, illustrate well the fulness sometimes considered desirable in such cases.§ But it will be observed that even here such important data as the ratios of expansion, release, and compression, and the quantity of cylinder condensation were not obtained.

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* This conclusion should in the view of the Author, be based on a limit of steam-pressure (perhaps above 75 lbs by gauge).

† For single cylinder or simple engines

‡ The Author questions the logic of this deduction

§ Bulletin de la Soc. Ind. de Mulhouse, Jan.-Feb. 1880. Journal Franklin Institute, Oct. 1885.

EXPERIMENTS ON A CONDENSING COMPOUND ENGINE, INDUSTRIAL SOCIETY OF MULHOUSE.

		July 8, 187[illegible]. Morning	July 8, 187[illegible]. Afternoon
Total Quantities	Duration of the experiments, in consecutive hours and decimals of an hour	3.00000	3.44139
	Total number of double strokes made by the pistons of the engine	16213.	17453
	Total number of pounds of feed-water pumped into the boiler	3248.068561	[illegible]
	Total number of pounds of condensing water admitted to the condenser	74038.567437	[illegible]
	Total number of pounds of condensing water and water of steam condensation withdrawn from the condenser	77253.139183	84407.777490
Engine	Steam, pressure in boiler, in pounds per square inch above the atmosphere	91.694	[illegible]
	Pressure in the condenser, in pounds per square inch above zero	1.8[illegible]	1.907
	Number of double strokes made per minute by the pistons of the engine	89.004448	88.711972
	Position of the throttle valve	Wide open.	Wide open.
	Fraction completed of the stroke of the piston of the small cylinder when the steam was cut off	0.25	0.42
	Fraction completed of the stroke of the piston of the small cylinder when the steam was released	0.98	0.98
	Fraction completed of the stroke of the piston of the small cylinder when the steam was cushioned.	0.925	0.925
	Fraction completed of the stroke of the piston of the large cylinder when the steam was cut off	0.45	0.45
	Fraction completed of the stroke of the piston of the large cylinder when the steam was released	0.91	0.91
	Fraction completed of the stroke of the piston of the large cylinder when the steam was cushioned	0.75	0.75
	Number of times the steam was expanded	9.6444	6.2569
	Atmospheric pressure, in pounds per square inch above zero	14.222813	14.222813
Water	Number of pounds of feed-water pumped into the boiler per hour	1081.888922	1315.914179
	Number of pounds of condensing water admitted to the condenser per hour	24661.273057	24648.018873
	Temperature in degrees Fahrenheit of the condensing water when admitted to the condenser	48.001	48.[illegible]
	Number of pounds of condensing water and water of steam condensation withdrawn from the condenser per hour	25732.004711	26068.394105
	Temperature in degrees Fahrenheit, of the condensing-water and water of steam-condensation when taken from condenser	86.937	[illegible]
	Number of Fahrenheit units of heat consumed per hour	1220557.870799	1471790.924665
Steam-pressures in Small Cylinder per Indicator	Pressure on piston of small cylinder at commencement of its stroke, in pounds per square inch above zero	99.787254	102.966053
	Pressure on piston of small cylinder at point of cutting off the steam, in pounds per square inch above zero	88.980990	91.454798
	Pressure on piston of small cylinder at the end of its stroke in pounds per square inch above zero	33.423600	45.184067
	Mean back pressure against piston of small cylinder during its stroke, in pounds per square inch above zero	35.343690	46.449021
	Back pressure against piston of small cylinder at the point where the cushioning began, in pounds per square inch above zero	34.846000	42.614000
	Indicated pressure on the piston of the small cylinder, in pounds per square inch	27.855379	30.678629
	Net pressure on the piston of the small cylinder, in pounds per square inch	24.227924	27.046174
	Total pressure on the piston of the small cylinder in pounds per square inch	60.200000	73.300000

		July 8, 1879. Morning.	July 8, 18[illegible], Afternoon.
Steam-pressures in Large Cylinder per Indicator.	Pressure on piston of large cylinder at commencement of its stroke, in pounds per square inch above zero.......	34.490000	43.[illegible]
	Pressure on piston of large cylinder at point of cutting off the steam, in pounds per square inch above zero....	21.156000	27.7[illegible]
	Pressure on piston of large cylinder at the end of its stroke, in pounds per square inch above zero	9.600000	12.5[illegible]
	Mean back-pressure against piston of large cylinder during its stroke, in pounds per square inch above zero..	3.756000	3.75[illegible]
	Back-pressure against piston of large cylinder at the point where the cushioning began, in pounds per square inch above zero.......	2.355000	2.3[illegible]
	Indicated pressure on the piston of the large cylinder, in pounds per square inch..............................	17.006220	22.3[illegible]
	Net pressure on the piston of the large cylinder, in pounds per square inch...............................	15.747273	21.24[illegible]
	Total pressure, in pounds per square inch above zero, on the annular surface of the piston of the large cylinder remaining after subtracting from the area of that piston the area of the piston of the small cylinder........	19.556220	24.8[illegible]
Horses-power.	Indicated horses-power developed in the small cylinder..	23.135964	25.11[illegible]
	Net horses-power developed in the small cylinder.... ..	20.123093	22.14[illegible]
	Total horses-power developed in the small cylinder......	50.000578	60.013[illegible]
	Total horses-power developed in the small cylinder by the expanded steam alone.....	28.101654	28.5[illegible]
	Indicated horses-power developed in the large cylinder..	40.738193	52.65[illegible]
	Net horses-power developed in the large cylinder..	37.722401	49.68[illegible]
	Total horses-power developed in the large cylinder by the annular surface of its piston remaining after subtracting from the area of that piston the area of the piston of the small cylinder	30.603792	38.33[illegible]
	Total horses-power developed in the large cylinder by the expanded steam alone	32.491077	40.194311
	Aggregate indicated horses-power developed by the engine	63.874157	77.771792
	Aggregate net horses-power developed by the engine....	57.845494	71.829041
	Aggregate total horses-power developed by the engine, exclusive of cushioning	80.604370	98.347430
	Aggregate total horses-power developed by the expanded steam alone in both cylinders inclusive of cushioning	60.598731	68.754356
	Horses-power developed by the engine at the friction brake applied to the wheel on the engine shaft	55.717593	67.499560
	Per centum of the total horses-power developed by the engine utilized as indicated horses-power............	79.24403	79.07862
	Per centum of the total horses-power developed by the engine utilized as net horses-power......	71.76471	73.03601
	Per centum of the total horses-power developed by the engine utilized as horses-power at the friction brake...	69.12477	68.63379
Economic Results.	Number of pounds of feed-water consumed per hour per indicated horse-power	16.937819	16.920179
	Number of pounds of feed-water consumed per hour per net horse-power	18.703080	18.320085
	Number of pounds of feed-water consumed per hour per total horse-power developed by the engine...........	13.422212	13.380260
	Number of pounds of feed-water consumed per hour per horse-power developed by the engine at the brake.....	19.417366	19.495149
	Number of Fahrenheit units of heat consumed per hour per indicated horse-power	19108.790286	18924.482608
	Number of Fahrenheit units of heat consumed per hour per net horse-power	21100.310264	20490.193161
	Number of Fahrenheit units of heat consumed per hour per total horse-power developed by the engine	1[illegible]142.576895	14965.219982
	Number of Fahrenheit units of heat consumed per hour per horse-power developed by the engine at the brake.	21906.148580	21804.449[illegible]

		July 8, 1879. Morning.	July 8, 1879. Afternoon.
... Indicator.	Number of pounds of steam present per hour in the small cylinder at the point of cutting off the steam calculated from the pressure there	607.871477	1005.726509
	Number of pounds of steam present per hour in the small cylinder at the end of the stroke of its piston, calculated from the pressure there ..	879.090774	1170.906554
	Number of pounds of steam condensed per hour in the small cylinder to furnish the heat transmuted into the total horses-power developed by the expanded steam alone in that cylinder	76.330424	78.900853
	Sum of the two immediately preceding quantities.	955.581198	1249.807407
	Number of pounds of steam present per hour in the large cylinder at the end of the stroke of its piston, calculated from the pressure there ...	805.995933	1047.498116
	Number of pounds of steam condensed per hour in the small and large cylinders, and in the receiver to furnish the heat transmuted into the total horses-power developed in those cylinders by the expanded steam alone after the closing of the cut-off valve on the small cylinder	162.065479	185.699642
	Sum of the two immediately preceding quantities	968.061412	1233.197758
... porized in the Boiler and the Weight of Steam accounted for by the Indicator	Difference in pounds per hour, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the small cylinder at the point of cutting off the steam	474.017445	310.187670
	Difference in per centum of the weight of water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the small cylinder at the point of cutting off the steam	43.813874	23.572029
	Difference in pounds per hour, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the small cylinder at the end of the stroke of its piston	126.307724	66.106772
	Difference in per centum of the weight of water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the small cylinder at the end of the stroke of its piston.......... ..	11.674740	5.023638
	Difference in pounds per hour, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the large cylinder at the end of the stroke of its piston	113.827510	82.716421
	Difference in per centum of the weight of water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the large cylinder at the end of the stroke of its piston	10.521183	6.285852
... to Large Cylinder alone.	Indicated pressure, in pounds per square inch, that would be on the piston of the large cylinder were the indicated pressure on the piston of the small cylinder divided by the ratio of the area of the small to that of the large cylinder, and the quotient added to the experimental indicated pressure on the piston of the large cylinder	26.664363	32.935300
	Net pressure in pounds per square inch, that would be on the piston of the large cylinder, similarly calculated to the immediately above	24.147684	30.418625
	Total pressure, in pounds per square inch above zero, that would be on the piston of the large cylinder corresponding to the total horses-power developed by the engine	33.648414	41.648801
	Back pressure, in pounds per square inch above zero, that would be against the piston of the large cylinder similarly calculated to the above ..	6.984051	8.713501

		July 8, 1879. Morning	July 8, 1879 Afternoon
Steam-pressures in Large Cylinder per Indicator	Pressure on piston of large cylinder at commencement of its stroke, in pounds per square inch above zero	34 490000	43 200000
	Pressure on piston of large cylinder at point of cutting off the steam, in pounds per square inch above zero	21 156000	27 740000
	Pressure on piston of large cylinder at the end of its stroke in pounds per square inch above zero	9 600000	12 561000
	Mean back pressure against piston of large cylinder during its stroke in pounds per square inch above zero	3 756000	3 756000
	Back pressure against piston of large cylinder at the point where the cushioning began, in pounds per square inch above zero	2 355000	2 355000
	Indicated pressure on the piston of the large cylinder, in pounds per square inch	17 006220	22 300000
	Net pressure on the piston of the large cylinder, in pounds per square inch	15 747273	21 041053
	Total pressure in pounds per square inch above zero, on the annular surface of the piston of the large cylinder remaining after subtracting from the area of that piston the area of the piston of the small cylinder	19 556220	24 850000
Horses-power	Indicated horses-power developed in the small cylinder	23 135964	25 113672
	Net horses-power developed in the small cylinder	20 12,09[illegible]	22 143736
	Total horses-power developed in the small cylinder	50 000578	60 013510
	Total horses-power developed in the small cylinder by the expanded steam alone	28 101654	28 560045
	Indicated horses-power developed in the large cylinder.	40 738195	52 658120
	Net horses-power developed in the large cylinder	37 722401	49 685305
	Total horses-power developed in the large cylinder by the annular surface of its piston remaining after subtracting from the area of that piston the area of the piston of the small cylinder	30 603792	38 333020
	Total horses-power developed in the large cylinder by the expanded steam alone	32 491077	40 194311
	Aggregate indicated horses-power developed by the engine	63 874157	77 771792
	Aggregate net horses power developed by the engine	57 845494	71 829041
	Aggregate total horses power developed by the engine, exclusive of cushioning	80 604370	98 347430
	Aggregate total horses-power developed by the expanded steam alone in both cylinders inclusive of cushioning	60 592731	68 754356
	Horses power developed by the engine at the friction brake applied to the wheel on the engine shaft	55 717593	67.499560
	Per centum of the total horses-power developed by the engine utilized as indicated horses-power	79 24403	79.07862
	Per centum of the total horses-power developed by the engine utilized as net horses-power	71.76471	73 03601
	Per centum of the total horses power developed by the engine utilized as horses-power at the friction brake..	69 12477	68.63379
Economic Results	Number of pounds of feed-water consumed per hour per indicated horse power	16 937819	16 900199
	Number of pounds of feed-water consumed per hour per net horse power	18.703080	18 320085
	Number of pounds of feed water consumed per hour per total horse power developed by the engine	13.422212	13 380260
	Number of pounds of feed water consumed per hour per horse power developed by the engine at the brake	19 417366	19 495149
	Number of Fahrenheit units of heat consumed per hour per indicated horse power	19108 790286	18924 482608
	Number of Fahrenheit units of heat consumed per hour per net horse power	21100 310264	20490 193261
	Number of Fahrenheit units of heat consumed per hour per total horse power developed by the engine	14142 576895	14965 219982
	Number of Fahrenheit units of heat consumed per hour per horse power developed by the engine at the brake	21906 148580	21804 449000

		July 8, 1879. Morning.	July 8, 1879. Afternoon.
Weight of Steam accounted for by the Indicator	Number of pounds of steam present per hour in the small cylinder at the point of cutting off the steam, calculated from the pressure there	607 871477	1005 726509
	Number of pounds of steam present per hour in the small cylinder at the end of the stroke of its piston, calculated from the pressure there	879 050774	1170 906554
	Number of pounds of steam condensed per hour in the small cylinder to furnish the heat transmuted into the total horses power developed by the expanded steam alone in that cylinder	76 530424	78 900853
	Sum of the two immediately preceding quantities	955 581198	1249 807407
	Number of pounds of steam present per hour in the large cylinder at the end of the stroke of its piston calculated from the pressure there	805 995933	1047 498116
	Number of pounds of steam condensed per hour in the small and large cylinders, and in the receiver to furnish the heat transmuted into the total horses power developed in those cylinders by the expanded steam alone after the closing of the cut-off valve on the small cylinder	162.065479	185 699642
	Sum of the two immediately preceding quantities	968 061412	1233 197758
Difference between the Weight of Water Vaporized in the Boiler and the Weight of Steam accounted for by the Indicator	Difference in pounds per hour, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the small cylinder at the point of cutting off the steam	474 017445	310 187670
	Difference in per centum of the weight of water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the small cylinder at the point of cutting off the steam	43 813874	23.572029
	Difference in pounds per hour, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the small cylinder at the end of the stroke of its piston	126.307724	66 106772
	Difference in per centum of the weight of water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the small cylinder at the end of the stroke of its piston	11 674740	5 023638
	Difference in pounds per hour, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the large cylinder at the end of the stroke of its piston	113,827310	82 716421
	Difference in per centum of the weight of water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the large cylinder at the end of the stroke of its piston	10.521163	6 285852
Cylinder-pressures Reduced to Large Cylinder alone	Indicated pressure, in pounds per square inch, that would be on the piston of the large cylinder were the indicated pressure on the piston of the small cylinder divided by the ratio of the area of the small to that of the large cylinder, and the quotient added to the experimental indicated pressure on the piston of the large cylinder	26.664363	32 935300
	Net pressure in pounds per square inch that would be on the piston of the large cylinder, similarly calculated to the immediately above	24 147684	30 418625
	Total pressure in pounds per square inch above zero, that would be on the piston of the large cylinder corresponding to the total horses-power developed by the engine	33 648414	41 648801
	Back pressure, in pounds per square inch above zero, that would be against the piston of the large cylinder similarly calculated to the above	6 984051	8 713501

This engine was two-cylinder compound, the cylinders placed side by side, an independent cut-off slide on the small cylinder, adjusted by a loaded governor, but with no cut-off valve on the low-pressure cylinder. The small cylinder was jacketed on its sides and the large cylinder half-jacketed. The dimensions were 112 and 18.9 inches (30 and 50 cm.), and 18.9 inches (50 cm.) stroke of piston.

In the table the "indicated" power is that shown by the diagrams: the "net" power is the indicated power less that expended in overcoming the friction of the engine; the "total" power is that shown by the indicator, including that expended in overcoming back-pressure; *i.e.*, it is given by the diagram measured down to the perfect vacuum line, and is the total mechanical effect produced by the fuel.

A trial of a small compound engine made under the direction of the Council of the Society of Arts, to determine its value as a motor for electric-lighting purposes, gave the following data.* The engine was of 5''.24 and 8''.98 diameter of cylinder, and of 14'' stroke of piston.

The data obtained from the log of the second trial being plotted, gave a graphical representation of the whole course of the trial, thus:

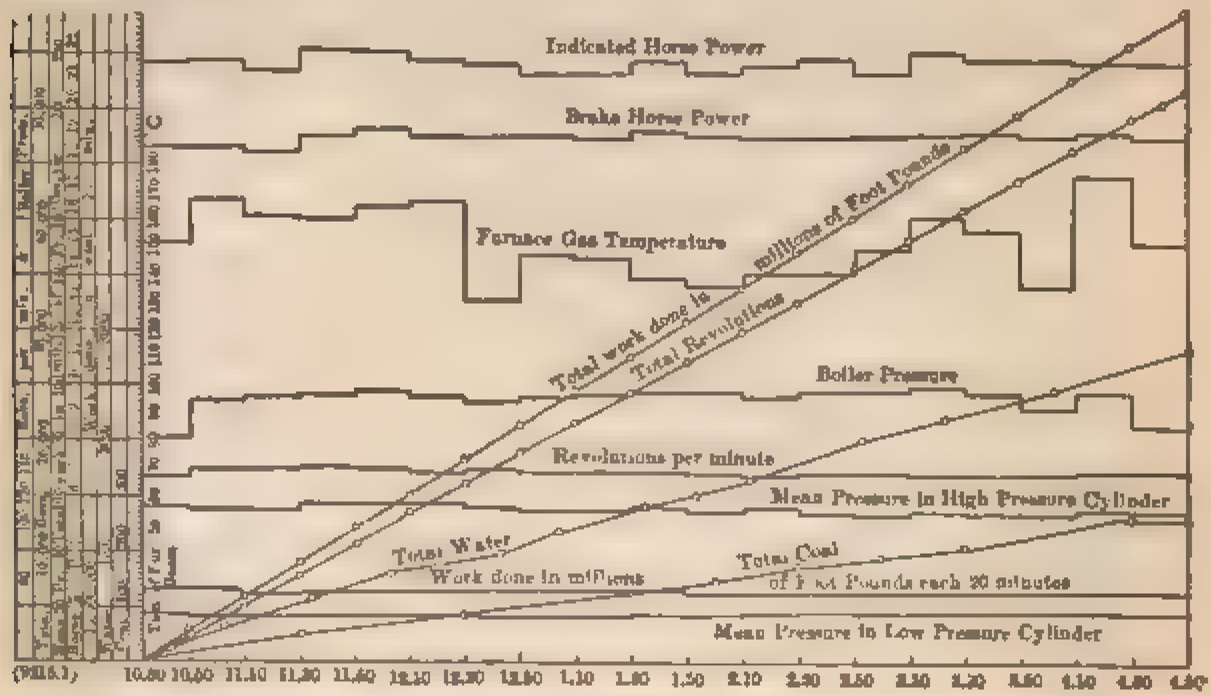


FIG. 121.—GRAPHICAL LOG.

* Journal Society of Arts, Feb. 15, 1889, p. 235.

Test of a Small Compound Engine.

1	Date	Sept. 26	Sept. 28	Oct. 1	Oct. 1
2	Trial	A1	A2	B	C
3	Duration	6.43 hours	6.27 hours	3.12 hours	½ hour
4	Power	Full	Full	Half	Empty
5	Revolutions per minute	140.48	137.35	138.10	144.20
6	Mean initial pressure, high-pressure cylinder	176.6	176.8	113.0	....
7	Mean ratio of expansion				
8	Mean effective pressure, high-pressure cylinder	52.93	54.98	33.25	11.32
9	Mean effective pressure, low-pressure cylinder	17.56	16.78	8.92	−.19
10	Indicated H. P. effective pressure, high-pressure cylinder	11.14	11.30	6.85	+2.40
11	Indicated H. P. effective pressure, low-pressure cylinder	10.98	10.26	5.47	−0.12
12	Indicated H. P. Total	22.12	21.56	12.32	2.28
13	Brake load net	288.8	288.0	147.6	....
14	Brake H. P.	19.44	18.95	9.76	....
15	Mechanical efficiency	0.879	0.879	0.792	....
16	Indicated H. P. in driving engine	2.68	2.61	2.56	2.28
	Boiler.				
17	Mean boiler-pressure (above atmosphere)	191.35	187.98	120.10	....
18	Atmospheric pressure for the day	14.9	14.8	14.8	....
19	Boiler-pressure (absolute)	206.25	202.78	134.9	....
20	Temperature of boiler-steam	384.3°	382.9°	350.1°	....
21	Pounds of feed used per hour	[448.7]	447.1	392.2	....
22	" " " indicated H. P. per hour	[20.28]	20.74	26.72	....
23	" " " brake H. P. per hour	[23.08]	23.59	33.73	....
24	Mean temperature of feed-water in tank	63.0	63.0	69.9	....
25	" " " before entering coil	[201]	201	....	....
26	Mean temperature of feed-water before entering exhaust feed-heater	115.7	63.0	....	....
27	Temperature of chimney gases (Fahr.)	355.4°	304.4°	369.1°	....
28	Coal per hour	39.66	40.70	27.25	....
29	" " " per indicated H. P.	1.79	1.89	2.21	....
30	" " " per brake H. P.	2.04	2.15	2.79	....
31	Pounds of water per pound of coal	....	10.99	12.08	....
32	" " " " from and at 212° F	....	11.71	12.76	....

Distribution of Heat A_2.

	Thermal Units.	Percentages.
Calorific value of 40.7 pounds of coal	577,900	100
Heat expended in heating and evaporating water, including heat given up by gases to coil in chimney	460,300	79.65
Heat expended in raising temperature of furnace-gases	40,700	7.05
Heat lost by radiation	51,070	8.85
Heat lost by imperfect combustion	15,450	2.68
Heat expended in evaporating moisture in coal	570	0.10
Heat lost in ash and otherwise unaccounted for	9,810	1.67
	577,900	100.00

Engine.—The work done, 21.55 indicated h. p., corresponds to 55,300 thermal units per hour, or 12.0 per cent. of the whole heat taken up by the water. The efficiency of a perfect heat-engine working between 383° and 212° F. would be 0.205. Such an engine receiving the same amount of heat as the Paxman engine, namely, 460,300 thermal units per hour, would turn into work 93,200 thermal units per hour. The actual efficiency of the engine therefore, compared with such a perfect engine, is 59 per cent. The heat received by the engine per indicated h. p. per hour was 21,350 thermal units. The brake h. p. of the engine was 18.95: its mechanical efficiency was therefore 87.9 per cent., the indicated h. p. expended in driving the engine itself being 2.61.

Boiler and Engine. The combined efficiency of the furnace, boiler, and engine, as represented by the consumption of fuel per horse power, works out to 9.6 per cent., 55,300 thermal units being turned into work per hour, with an expenditure of fuel having a value of 577,900 thermal units. The coal used per indicated h. p. per hour was 1.89 pounds, and per brake h. p. per hour 2.15 pounds.

Steam per Indicator-cards.—The amount of steam shown by the indicator-diagrams (Figs. 18 and 19) was as below:

	Percentage of whole weighed feed-water.
Steam in h. p. cylinder at a pressure of 150 lbs per square inch above the atmosphere, which corresponds to a point at 0.39 of the stroke, a little after cut-off in all cases.....	65.0
Steam in l. p. cylinder at a pressure of 10 lbs. per square inch above the atmosphere, which corresponds to a point at 0.67 of the stroke, well before release in all cases.. ..	78.8

96. **Examples of Trials of Portable Engines** are, perhaps, best found in the annual reports of the competitive tests of such engines at the various agricultural exhibitions, where they are most frequently seen and subjected to trial. These trials are of peculiar interest, as exhibiting the fact that even the smallest engines may, by skilful design, construction, and management, be made to give admirable economy. The following are the data and the results reported at Newcastle, G. B., by the committee of judges, aided by Messrs. Sir Frederick Bramwell and W. Anderson: *

* Journal Agricult Soc of England, vol xxiii 1887

Dimensions of Engines.

	Simple Engines.						Compound Engines				
Catalogue number	**8111**	**8114**	**147**	**8108**	**8125**	**8117**	**8115**	**8118**	**8124**	**8107**	**8116**
Diameter and stroke of high-pressure cylinder in inches	8½×12	7¼×10	5¾×9	8¼×15	9¼×12	10½×14	6×11	4¾×10	5¾×14	5¾×15	7½×14
Diameter and stroke of low-pressure cylinder in inches	....	....	....	....	....	....	9×11	9¼×10	9¾×14	9×15	12×14
Mean clearances and ports, high pressure cylinder, in cubic inches	100.7	....	23.0	99.5	90	....	75.0	....	54.0	62.2	
Mean clearances and ports, low-pressure cylinder in cubic inches	....	....	....	....	....	....	168.3	....	128.5	103.0	
Intermediate space cubic inches							875		517	427.4	
Diameter and width of fly wheel	48½×6¼	53×6	40×5	60×7½	60×7	60×7½	48×6¼	53×6	60×6	60×7½	60×7½

Principal Dimensions of Boilers

	8111	8114	147	8108	8125	8117	8115	8118	8124	8107	8116
General description of Boiler	Loco	Loco.	Vertical	Loco	Loco	Loco	Loco	Loco	Loco	Loco.	Loco
Firebox and Grate											
Area of ordinary grate in sq. feet	6.68	3.35	2.6	6.7	5.73	5.93	3.67	3.35	5.33	6.7	5.93
Area of grate as used at trial in sq. feet	5.98	2.63	2.6	3.39	4.69	4.18	3.67	2.63	4.32	3.39	4.18
Width of bars in inches	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Width of air spaces in inches	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Area of air spaces in sq. feet	2.2	.96	.51	1.98	1.30	2.17	.75	.96	1.24	1.98	2.17
Area of air-spaces as used at trial in sq. feet	1.7	.77	.51	1.11	1.16	1.53	.75	.77	1.1	1.11	1.53
Height of firebox crown above bars in inches	36¼	30¾	24¾	30¾	32¼	29	31½	25¾	31	30¾	29
Heating Surface											
Length from out to out of tube-plates in inches	75	72	35	81¾	84¾	79¾	82½	66	84¾	81¾	79¾
Number of tubes	24	76	27	51	53	30	22	76	53	51	30
Material of tubes	Iron	Steel	Iron	Iron	Steel	Iron	Iron	Steel	Steel	Iron	Iron
Outside diameter of tubes in inches	2½	1½	1¾	2	2	2½	2¼	1½	2	2	2½
Inside diameter of tubes in inches	2¼	1¼	1½	1¾	1¾	2¼	2	1¼	1¾	1¾	2¼
Heating surface of firebox in sq. feet	37.2	20.45	13.8	34.5	*40.8	28.3	23.0	17.0	28.1	34.5	28.3
Heating surface of tubes in sq. feet	106.9	188.5	35.2	180.1	194.2	135.6	98.3	172.6	194.2	180.1	135.6
Heating surface of smokebox in sq. feet	4.59	2.6		3.5	3.1	3.8	3.6	2.6	4.4	3.5	3.8
Total water heating surface in sq. feet	148.7	211.5	49.0	218.1	238.1	167.7	124.9	192.2	226.7	218.1	167.7
Sq. feet of water-heating surface to nominal H. P. of engine	18.6	26.4	16.3	27.2	29.6	20.9	15.6	24.0	28.3	27.2	20.9
Ratio of area through tubes to normal grate-area	.122	.274	.177	.092	.164	.171	.171	.274	.176	.092	.171

* This includes 8.37 square feet of heating-surface, due to the eight Paxman water tubes in the firebox.

[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Arrangement of uptake damper	Hinged uptake door.	No damper in uptake, throttle valve in chimney	Hinged uptake door.	Hinged uptake door.	Hinged uptake door.	Hinged uptake door.	Hinged uptake door.	Damper in uptake, throttle valve in chimney	[illegible] uptake door	[illegible]

Results of Experiments connected with the Boilers.

(Theoretical evaporative power of the coal from and at 212° = 15.45 lbs. Air theoretically required per 1 lb. of coal 11.38 lbs.)

1 Water supplied to engines lbs.	[illegible]	1411	[illegible]	1868	[illegible]	2804	2414	1394	1658	1967
2 Temperature of water supplied °F.	60	61	[illegible]	61	63	62	63	65	64	63
3 Average temperature to which the feed water was raised by condensed steam °F.		85.6	...	91.6	123.6	104.5	154.3	82.5	118.2	91.4
4 Calculated weight of steam condensed in heating feed water lbs.		96.6	...	49.10	96.1	114.5	215	[illegible]	84.7	51.4
5 Calculated weight of steam condensed in jacket per from and at 212° lbs.		161.7	..	272.4	290.5	214.8	....	215	269.8	280.4
6 Coal consumed lbs.	[illegible]	148.25	[illegible]	199.75	193	351	259	148.5	168	202.5
7 Water evaporated from and at 212° exclusive of jacket water lbs.	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	3343	2786	1606	1912	2270
8 Water as above evaporated per 1 lb. of coal	[illegible]	[illegible]	6.80	10.79	10.01	9.52	10.75	10.81	11.38	11.21
9 Estimated weight of water evaporated including that from jackets from and at 212° lbs.		[illegible]	...	2408	[illegible]	3337.8	....	1821	2181.7	2550.4
10 Water as above evaporated per 1 lb. coal from and at 212° lbs.		[illegible]	...	12.07	11.21	10.14	....	12.96	12.90	12.59

* This includes [illegible] square feet of heating surface, due to the eight Paxman water tubes in the firebox.

RESULTS OF EXPERIMENTS CONNECTED WITH THE BOILERS.—*Continued.*

	Simple Engines.						Compound Engines.				
Catalogue number	8111	8114	147	8108	8128	8117	8115	8118	8124	8107	8110
11. Efficiency of boiler and of its management, taking the theoretical value of the coal at 15.45	.589	.839	.442	.786	.725	.636	.696	.794	.840	.814	
12. Weight of air passed through per 1 lb of coal burnt .... lbs.	23 17	22 48	16 91	26 20	23 45	17 435	10 75	13 20	24 41	27 43	
13. Percentage of air in relation to theoretical quantity 11.38 lbs. per 1 lb. .... %	203.6	109.1	148 6	230 3	206 2	153.2	167 0	130 7	214 7	241 0	

RESULTS OF EXPERIMENTS RELATING TO THE HORSE-POWER AND THE COAL-CONSUMPTION OF THE ENGINES.

	8111	8114	147	8108	8128	8117	8115	8118	8124	8107	8110
1. Number of brake used	3	2	2	3	1	3	3	2	1	1	3
2. Time running .... minutes	245	263 5	251	263	263	248	245	262	268	264	148
3. Number of revolutions .... total	37,342	45,400	39,178	35,482	35,550	35 [illegible]	42,436	41,730	37,490	30,340	26,940
4. Revolutions per minute	152 4	172 3	156 1	136 3	135 2	145 9	172 9	159 2	139 9	149 [illegible]	181 8
5. Declared number of revolutions per minute	150	168	160	130	132	130	170	156	134	139	189
6. Total weight hanging on brakes .... lbs.	187 75	120 75	43.5	233 75	236 5	210.75	186 75	202.75	274 2	265 75	122 75
7. Circumference of brake load circle .... feet	17 32	17 565	17 28	17 31	17 295	17 3	17 31	17 575	17 24	17.31	17.31
8. Coal consumed during runs .... lbs.	404	138 25	113	199 75	193	151	249	148.5	168	202 5	249
9. Coal consumed per hour .... lbs.	98 93	31 48	27 01	45 57	44 03	37 326	61 45	34.24	37.61	46.00	100 1
10. Indicated horse-power	....	13 88	*3 508	23 43	19 80	20 15	21 12	18 61	22 77	24.00	22.1
11. Declared horse-power	16	12	4 5	17	17	16	18	18	20	20	20
12. Brake horse-power, excluding correction for friction of brakes	15 02	11 08	3 555	16 71	16 69	16 80	16 93	17.87	20.16	20.77	18.29
13. Brake horse-power, with friction of brakes added	15 31	11 37	3 798	16 976	16 94	17 08	17.25	19.47	20.33	21.07	18.50
14. Coal consumed per hour per brake horse-power, excluding friction of brakes .... lbs.	6 588	2 840	7 6	2 727	2 638	5 20	3 746	1.977	1.874	2.267	5.694
15. Coal consumed per hour per brake horse-power, with friction of brakes added .... lbs.	6 466	2 766	7 111	2 684	2 590	5 124	3.674	1.941	1.850	2.184	5 418
16. Horse power as calculated at the time of the trials	16 19	12 27	4.389	17 83	17 46	17 99	15.90	18 40	20 80	22.00	19.84
17. Coal consumed per brake horse-power as calculated at trials, and published in the papers .... lbs.	6 11	2 566	6 030	2.555	2 52	4 89	3.47	1.847	1.80	2.09	5.24
18. Ratio of indicated horse-power to brake horse-power, with friction allowance, as line 13.	....	.819	*1 080	.774	.859	.848	.818	.943	.891	.877	.837

* The indicator-pipes supplied to this engine by the maker were very small.

COMPARISON OF COAL CONSUMPTION BETWEEN THE CARDIFF AND NEWCASTLE TRIALS.

	Simple Engines.						Compound Engines.		
	Newcastle Trials.			Cardiff Trials.			Newcastle Trials.		
Catalogue Number	8125	8108	8114	4942	2927	4942	8124	8118	8107
	Powell's Duffryn Coal.			Llangennech Coal.			Powell's Duffryn Coal.		
Coal consumed per brake horse-power per hour, without allowance for friction of brakes	2.638	2.727	2.842	2.79	2.881	2.884	1.874	1.977	2.267
	Powell's Duffryn reduced to Llangennech.						Powell's Duffryn reduced to Llangennech.		
Consumption of coal as above, corrected for superior quality of Powell's Duffryn used at Newcastle	2.678	2.768	2.884	2.79	2.881	2.884	1.902	2.007	2.301
	Powell's Duffryn Coal.						Powell's Duffryn Coal.		
Coal consumed per brake horse-power per hour, with allowance for friction of brakes	2.599	2.684	2.766	....	....	....	1.850	1.943	2 184

97. The General Conclusions deducible from this remarkably full and accurate collection of data are correspondingly interesting and important. Classifying the data, computing the quantities of heat, and constructing a balance-sheet, the following table was obtained as illustrating the operation of the prize engine No. 3125:

BALANCE-SHEET OF SIMPLE ENGINE No. 3125.

DR.	Units.	CR.	Units.	Percentage.
To the heat developed in furnace—		By heat expended:		
In the combustion of wood:		1. In evaporating the water in the wood and heating its steam to 385°	9,557	.32
From Carbon..........	79 331	2. In heating the wood and the air required for its combustion from 70° to 385°..........	3,884	.13
" Hydrogen........	4,816			
In the combustion of coal:		3. In evaporating the water in the coal and heating its steam to 385°....	8,374	.29
From Carbon..........	2,481,400			
" Hydrogen........	337.475	4. In heating the coal and the air required for its combustion from 70° to 385°..........	129,321	4.44
" Sulphur.........	9,071			
		5. In displacing the atmosphere by the products of combustion of the wood and the coal with the air needed for their combustion......	53,394	1.83
		6. In heating the excess of air	130,980	6.34
		In displacing the atmosphere by the excess of air	53,509	
		7. In evaporating the water in the boiler.	2,090,300	71.78
		8. In radiation and convection	271,307	9.32
		9. In ashes and unconsumed fuel......	53,915	1.85
		10. Unaccounted for..................	107,552	3.70
	2,912,093		2,912,093	100.00

phere for its reception, gives a further amount of loss = 6.34 per cent., which is preventable. In the case of this engine, No. 3125, we have an excess of air weighing 12.31 lbs. for each lb. of fuel burnt, being practically equal to the air which is theoretically needed,—while in engine No. 3114 the excess was only 1.67 lbs., and in engine No. 3113, 4.02 lbs. It is clear, therefore, that if 3125 had been worked with no greater excess than 3114, the 6.34 per cent. of loss of item 6 would have been reduced by 5.49 per cent., leaving only .85 per cent.

Analysis by Messrs. Pattinson and Stead, Middlesbrough, of Powell-Duffryn's Coal used at the Newcastle Trials.

Samples were collected at intervals during the trials, and the coal analyzed was an average of these:

Carbon, . . .	88.40	available
Hydrogen, . .	3.65	$- 0.32 = 3.33H$
Oxygen, . . .	2.55	$0.32H = 2.87H_2O$ (water)
Nitrogen, . . .	0.64	
Sulphur, . . .	0.76	= 1.36 per cent. pyrites
Ash, ...	3.17	
Water, ...	0.83	
	100.00	
Sulphur in ash, .	0.04	

Calorific value in British Thermal Units.

Carbon, . . .	.884 × 14,544 units	=	12,856 units
Hydrogen, .	.0333 × 61,200 "	=	2,037 "
Pyrites estimated at	. . . .		47 "

Total per one pound of coal, . . . 14.930 units

Weight of air required to burn one pound of coal, 11.38 lbs.

98. Trials of Locomotive Engines are more difficult of prosecution than are those of any other class of steam-engine. The conditions of its operation are such that it is very difficult to secure measurements of coal and of water consump-

tion, exceedingly difficult to obtain a good arrangement for taking indicator-diagrams, and next to impossible to determine the quality of steam made while in regular work. It is commonly expected that the engineer conducting the trials will report on the following points:

(1) The dimensions of the engines, as to diameter of cylinder, stroke, diameter of boilers, exhaust-nozzles, etc.

(2) Their weight and the distribution upon driving-wheels.

(3) The weight of the train hauled.

(4) The weight of coal consumed in hauling the same train over the same route by each engine.

(5) The quantity of water evaporated by each engine in doing the work.

(6) The relative amounts of smoke and cinders which escaped from the smoke-stacks.

(7) The temperature of the gases escaping.

(8) The tractive resistance of the train at the same places as indicated by a dynamometer during the trips of each engine.

(9) The pressure of steam in the cylinders of the several engines, as shown by indicator-diagrams, the pressure indicated in the boilers being recorded at the same time.

(10) The time occupied in making each trip and between points, which should be as nearly uniform as possible.

(11) The temperature of the air at starting, in the middle, and at the end of each trip.

(12) The temperature of the water in the tenders.

Such trials should usually be made as nearly as possible under the ordinary conditions of every-day work.

The engines are weighed in working trim, with steam up and ready to start, and also with steam off and water blown out. The coal is weighed by taking the weight of tender empty and loaded, and noting the difference in weight as that of coal consumed on the run, returning the balance at its end. The water is commonly measured by using a float in the tank, the rise and fall of which, and the area of water-surface, being known, the volume and weight of water become easily determinable. The water may also be weighed, as well as the coal. The rela-

tive quantities of smoke and cinder ejected can only be estimated from observation. The engine should be given a preliminary provisional trial to see that everything is in working order.

The following is an abstract of a good example of a report describing work of this kind under the direction of Mr. Leavitt, the trial being conducted by Professor Coon:

The object of these tests was to determine the relative efficiency of Strong's locomotive boiler and cylinder and valve gear, and their ease of working and liability to derangement, as compared with the best type of American locomotive in common practice at the present time. To this end three locomotives were tested, viz.: Engine No. 444, fitted with Strong's twin-furnace boiler and a four-valve cylinder and valve gear; engine No. 383, having an ordinary straight-top boiler, with fire-box over instead of between the frames, anthracite-coal grates, and fitted with cylinder and valve gear similar to No. 444; engine No. 357, having an ordinary boiler similar to that on engine No. 383, save that it has a "wagon top" of eight inches, and the link-motion common in American practice, with plain slide-valves having a good balancing device.

The leading particulars of the three locomotives are as follows:

	Engine 444.	Engine 383.	Engine 357.
Cylinders, etc.			
Cylinder, diameter and stroke	20 ins. by 24 ins.	19 ins. by 24 ins.	20½ by 24 ins.
Diameter of piston-rod	3¾ ins.	3½ ins.	2⅞ and 2¾ ins.
Length of connecting-rod, centres	8 ft. 3 ins.		
Transverse distance between cylinder, centres	7 ft.		
Distance from centre of main drivers to centres of cylinders	12 ft. 10 ins.		
Number of valves per cylinder	4, 2 steam, 2 exh.	4, 2 steam, 2 exh.	One
Type of valves	Gridiron	Gridiron	Balanced slide.
Number of ports per valve	2	10	
Size of ports	4⅞ ins. by ⅞ in.		
Full travel of valves	$1\frac{1}{16}$ in.	$1\frac{1}{16}$ in.	
Lap of valves	$\frac{1}{16}$ in.		
Lead of valves, steam	⅛ in. constant	⅛ in. constant	
Lead of valves, exhaust	$\frac{1}{16}$ in. constant	$\frac{1}{16}$ in. constant.	
Throw of eccentrics	2¼ ins.		
Tractive force per lb. of mean effective pressure on piston	154.8 pounds	131.3 pounds	149.11 pounds.
Cylinder clearance in cubic inches	481	448	568.
Cylinder clearance in per cent. of piston displacement	6.38 per cent.	6.58 per cent.	7.35 per cent.

	Engine [illegible]	Engine [illegible]	Engine [illegible]
Wheels and Journals.			
Driving and truck wheel centres [illegible]	Wrought-iron	Cast-iron	Cast-iron
Front truck 4-wheeled swing beam	Yes	Yes	Yes
Rear truck 2-wheeled swing and radius bar	Yes	None	None
Nominal diameter of driving-wheels	[illegible] ins	[illegible] ins	56 ins.
Callipered diameter of driving-wheels	[illegible] ins	[illegible] ins	[illegible] ins.
Diameter of front truck-wheels	30 ins		
Diameter of rear truck wheels	42 ins		
Total wheel-base of engine	[illegible] ft. [illegible] ins.	[illegible] ft. [illegible] in	[illegible] ft. 1 in.
Rigid wheel-base of engine	[illegible] ft. [illegible] ins	[illegible] ft. [illegible] in	[illegible] ft.
Driving axle-journals (diameter and length)	[illegible] in. by 12 ins		
Front truck axle journals (diameter and length)	6 ins. by 11 ins.		
Rear truck axle journals (diameter and length)	[illegible] ins. by [illegible] ins		
Main crank-pin journals (diameter and length)	[illegible] ins. by [illegible] ins		
Coupling-rod journals (diameter and length)	4½ ins. by 4 ins.		
Weights, etc.			
Weight on first pair drivers, in working order	[illegible],000 pounds		
Weight on second pair drivers, in working order	30,000 "		
Weight on third pair drivers, in working order	30,000 "	None	None.
Total on drivers, in working order	[illegible] "	74,640 pounds	63,280 pounds.
Weight on front truck, in working order	27,000 "	24,880 "	27,440 "
Weight on rear truck, in working order	21,000 "	None.	None.
Total weight of engine, in working order	138,000 "	99,520 pounds	90,720 pounds.
Boilers, etc.			
Height of boiler-centre above rail	7 ft. 3 ins.		
Kind of boiler	Twin-furnace	Straight-shell	Wagon-top.
Material for boiler-plate	O. H. steel	O. H. steel	O. H. steel.
Diameter of barrel inside smallest ring	58 ins.	55 ins.	54 ins.
Diameter of fire-boxes and combustion chambers, corrugated	38¼ ins. inside		
Length of grates	42⅞ ins. outside	11 ft	11 ft.
Width of grates		3 ft. 4½ ins.	3 ft. 6⅝ ins.
Number of tubes, all iron	306	[illegible]	248.
Diameter of tubes, outside	1¾ ins	2 ins	2 ins.
Length of tubes	11 ft. 5 ins.	11 ft. 4½ ins.	12 ft. [illegible] ins.
Grate area, square feet	62 square ft.	37.12 square ft.	[illegible] square ft.
Heating surface, fire-box, square feet	155	151.6	148.3.
Heating surface, combustion chamber, square feet	93		
Heating surface, tubes, square feet	1600	1234.3	1429.8.
Heating surface, total, square feet	1848	1385.9	1574.1.
Ratio of heating-surface to grate area	29.8 to 1	37.3 to 1	40.1 to 1.
Smallest inside diameter of smoke-stack	16 ins.		
Height of top of smoke-stack above rail	14 ft. 3 ins.		
Working steam-pressure per sq. inch	160 lbs	160 lbs.	140 pounds.
Tender.			
Eight-wheeled, double trucks (diameter of wheels)	33 ins	33 ins	33 ins.
Capacity of tender (gallons)	3000	3000	[illegible]
Capacity of tender (coal)	10,000 pounds	10,000 pounds	10,000 pounds.
Weight of tender loaded	70,000 "	70,000 "	[illegible] "
(Same tender used on all tests.)			

All the locomotives were subjected to exactly the same work, under similar conditions. The route was a continuous succession of curves as sharp as 14 degrees, and grades as steep as 96 feet per mile.

The water supplied to the tender on each trip was gauged as follows: On each side of the tender, and at the centre of gravity of the water space, long glass gauges, the height of the tender, were attached, with a blank wooden scale behind. The tender was filled nearly full of water and placed on the track scales, the height of water in the glass gauges being marked on the scale. These were the 0 marks. The water was then drawn off five cwt. at a time (560 pounds), and corresponding marks made on the scale, each division thus representing 560 pounds of water. The readings of both scales were taken immediately before and after taking water, and at the end of the run. The same tender (that of engine No. 444) was used on all the trials, without alteration.

All the coal for each run was weighed in barrows on a platform scale, and any coal remaining in the tender at the end of the run was deducted from the original amount, it also being weighed. At the beginning of each run, before any coal was charged to the experiment, the engine was allowed to start with a uniformly good fire. It was not practicable to get the weight of the ashes.

The boiler-pressure was taken at five-minute intervals on each run.

The temperature of the feed-water at the three tanks where water was taken did not vary half a degree from 64.3° F. throughout the trials.

Engine No. 357 has a boiler feed-pump attached to one cross-head, any deficiency of water being supplied by an injector. Engines Nos. 444 and 383 were supplied with water solely by injectors.

All indicator-cards were taken on up-grade. Two observers took simultaneous cards from each end of both cylinders, while a third, in the cab, noted the steam-pressure, position of throttle-lever and reverse-lever, the exact time the cards were taken,

and the exact time of passing the mile-posts, and the time of stopping and starting at stations.

The indicator-rig, for reducing the piston-motion for the indicator-barrel, was a modified true pantagraph motion. The strings used were hard-braided linen, about 8 inches long. Cards taken at 60 miles per hour are not more than .03 inch longer than those taken at one mile per hour. No wind-shields were used, and not the slightest difficulty was experienced in taking accurate cards at over 60 miles per hour.

Through the courtesy of the Pennsylvania Railroad, their dynamometer car was used.

Attention is invited to the tables of coal consumed and water evaporated. On the first four trips with Engine 444 some journals on the front truck heated, and on the third trip a main crank-pin heated, so that a helper was considered desirable. Subsequently no parts warmed at all on any of the engines.

If the consumption of coal of the three locomotives be compared, each on two trips when they were using the same grade of coal, to wit: Locomotive No. 444, on trips No. 4 and No. 10; locomotive No. 383, on trips No. 8 and No. 14; and locomotive No. 357, on trips No. 12 and No. 13,—it gives the average for the three respectively, as follows:

Average for engine No. 444, . .	6,537	pounds coal
" " " " 383, . .	7,441	"
" " " " 357, . .	8,087	" .

—which is a difference of 646 pounds between Nos. 383 and 357, or an advantage of 8.7 per cent. in favor of engine No. 383, and a difference of 1550 pounds between Nos. 444 and 357, or an advantage of 23.7 per cent. in favor of No. 444. It is also to be borne in mind that engine No. 357 has 2 square feet more grate area and nearly 200 square feet more heating-surface and much better steam room than No. 383. With equal boilers there would be still greater difference in the coal. In support of this, compare the two runs of engine No. 383 on May 2 and May 10, and engine No. 357 on May 7 and May 9,

In the following table, column 1 gives the per cent. which the weight of engine and tender is of the weight of the train, while column 2 gives the per cent. which traction due to engine and tender is of traction due to train:

	1.	2
Sugar Notch to Fairview, May 19, ...	69.6	80.0
" " " 20, ...	50.0	54.0
Mauch Chunk to Penn Haven Junction, May 19,	42.5	63.0
Penn Haven Junction to White Haven, May 19,	42.5	62.9
White Haven to Glen Summit, May 19, . . .	42.5	45.1

Strain-diagram A fairly represents the action of this locomotive. The portion shown was taken on a 96-foot grade and on a 10-degree curve, and at a speed of 13.1 miles per hour, when the locomotive was working at its best. Strain-diagram B is a portion of that taken on May 20, with engine No. 444, the portion shown being taken at precisely the same portion of the road as diagram A, viz.: on 96-foot grade and 10-degree curve, but at a speed of 27.2 miles per hour. The revolutions of the drivers of engine No. 82 can be accurately taken from the strain-diagram, every cusp on either side of the curve representing a revolution.

It is obvious from this diagram (A) that one end of one cylinder was doing more work than the other end of the other cylinder. Hence the great oscillations in the diagram. It is also to be noted that the train, as a whole, had its rate of oscillation about every six or seven revolutions of the drivers.

Speed of trains:

	Miles per Hour.
Engine No. 82, ...	13.1
Engine No. 444, ...	27.2

Net tractive force at time diagrams A and B were taken:

	Pounds.
For engine No. 82, ...	15,097
For engine No. 444, ...	8,718

Resistance or tractive force, in pounds, per long ton:

For engine No. 82 (diagram A), ...	53.5
For engine No. 444 (diagram B), ...	50.1

Cylinders, 20 in. x 26 in.; drivers, 50 in. diameter; weight of train, 281.7 long tons; 96-foot grade; 10-degree curve; speed, 13.1 miles per hour; net horse-power, 527.4; net tractive force, 15,097 pounds; tractive force per ton, 53.5 pounds.

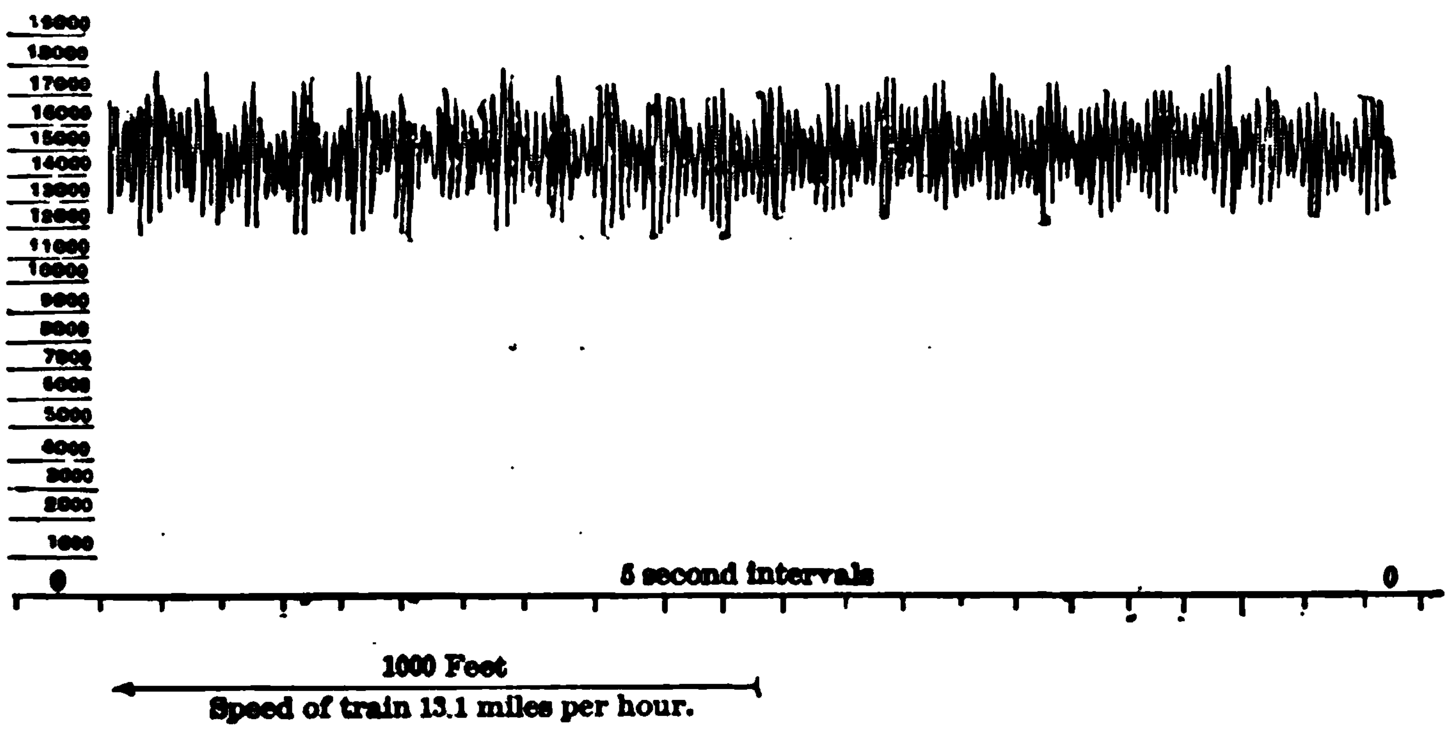


FIG. 122.—STRAIN-DIAGRAM "A," LOCOMOTIVE NO. 82.

Weight of train, 147 long tons; 96-foot grade; 10-degree curve; speed, 27.2 miles per hour; indicated horse-power, 952.4; net horse-power, 632.3; net tractive force, 8718 pounds; tractive force per ton, 50.1 pounds.

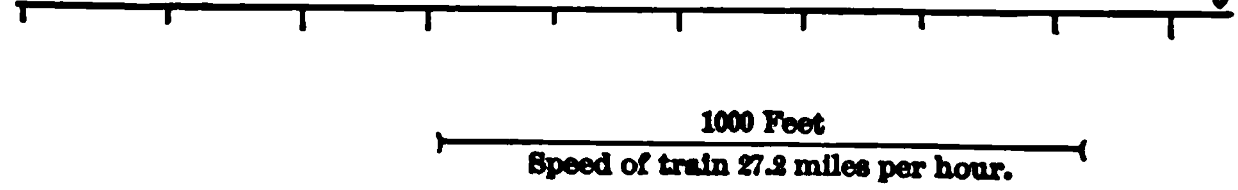


FIG. 123.—STRAIN-DIAGRAM "B," LOCOMOTIVE NO. 444.

For the above strain-diagrams, A and B, the net horse-power, due to tractive force of 15,097 pounds and 8718 pounds for the respective speeds are: For engine No. 82, net horse-power, 527.4; for engine No. 444, net horse-power, 632.3; or the net

horse-power of engine No. 82 was 83.4 per cent. of net horse-power of engine No. 444.

The indicated horse-power at the time diagram B was taken is as follows:

Mean effective pressures, in pounds, per square inch:

	P. R. End.	Front End.
Right-hand cylinder, ...	88.64	85.84
Left-hand "	90.21	82.48
Total, ...	178.85	168.32

Indicated horse-power, P. R. end, ...	484.3
" " front "	468.1
Total indicated horse-power, ...	952.4

Horse-power due to traction of No. 444 engine and tender, $952.4 - 632.3 = 320.1$. Per cent. which traction of No. 444 engine and tender was of traction of train, 50.6 per cent. Per cent. which weight of No. 444 engine and tender was of weight of train, 50.0 per cent.

As the weight of train drawn by engine No. 444 was in part estimated, it is probably not correct within 0.6 per cent. The weight of train drawn by engine No. 82 was obtained exactly.

The following is a *résumé* of experiments on locomotives, made by M. Marié, engineer of the Paris and Lyons Railroad, on the heavy grades leading to the Mont Cenis Tunnel, which may be compared with the preceding:

Water evaporated by pound of coal, ...	8.88
Consumption of coal per indicated horse-power,	2.88
" " " effective " .	3.27
Average speed per hour during trials, ...	17.04
Efficiency of boiler, ...	65 per cent.
Efficiency of engine, as compared to "perfect" engine working under the same range of temperatures, . . .	53 "

Dimensions of Locomotive.

Cylinders, $21\frac{1}{2} \times 26$ in. stroke. Drivers, $49\frac{5}{8}$ in. diam.

Heating-surface—fire-box, ...	104 sq. ft.
" —tubes	2,045 "
Total, ...	2,149 sq. ft.
Grate area, ...	22.4 "
Weight of engine and train, ...	366,474 lbs.

The coal was of good quality, yielding 14,600 British thermal units when burnt in oxygen. The amount of ash was 6.5 per cent., and the coal contained 1 per cent. of moisture. The average point of cut-off during the experiments was at 19 per cent. of the stroke.

99. **The Several Results and Conclusions** to be derived from these, as from locomotive-trials generally, are very obvious. The comparatively small quantity evaporated by the fuel is evidence that the necessarily large amount of work demanded of the locomotive boiler is obtained at great sacrifice of efficiency. Much larger figures than those above given are often quoted, as, for example, in the French report referred to above; but it is probably generally the fact that some priming has produced a misleading result.* The second important matter is the consumption of water, an especially serious item in the performance of locomotives. In the trials above detailed, the weight of water used ranged from 24 to 40 pounds per I. H. P. per hour. The latter is a fair result; the former is remarkably good. The coal account gives from 3 to 5 pounds per I. H. P. per hour. Four pounds seems to be a good average amount. Little need be added to what has already been said in reference to the data and the deductions from the dynamometer records. The traction ranged, in the case given, in the neighborhood of 50 pounds per ton, or about 2.2 per cent.; and the power exerted was from 600 to nearly 1000 horse-power.

The results reported from the French trials quoted are re-

* See Clarke's Manual, p. 799.

markably and exceptionally excellent. Later constructions of compound locomotives, only, have rivalled them. It will be noted that the boiler had very high efficiency, 65 per cent., and that the engine had a total efficiency above one-half that of the ideal, perfect, engine operating under similar external conditions.

The general results of comparisons of performance of simple and compound locomotive engines show an evident gain by the latter, to the extent of from 15 to 20 per cent., in fuel and steam consumption; which gain is often partly compensated by a somewhat greater consumption of oil. The saving in coal consumed is often less than that of steam used; the two quantities being, in some reported cases, 15 and 20 per cent. respectively. The pressure of steam adopted in these comparisons is commonly 10 to 12 atmospheres.

100. Trials of Marine Engines are conducted under somewhat more favorable conditions than attend trials of locomotives, but they are also to some extent embarrassed by the peculiar surroundings of the motor apparatus. Some of the most interesting and fruitful investigations ever made have, notwithstanding these difficulties, been made in this direction. The now well-known trials of the U. S. R. M. steamers, designed by Mr. Emery, are examples of such, and their reported performance is here presented.*

The "Rush," the "Dexter," and the "Dallas" are similar as respects the hulls, the screws, and the boilers; but the engines are different: that of the "Rush" being compound; that of the "Dexter," high-pressure condensing; and that of the "Dallas," low-pressure condensing. The vessels are each 129½ feet between perpendiculars at water-line, 23 feet extreme breadth of beam, and 10 feet depth of hold. The draught of water aft is about 8 feet 10 inches. The hulls are of wood. One of the vessels averaged upward of eleven nautical miles per hour for six consecutive hours on her trial trip, and neither of them averaged less than ten knots.

Each vessel has one boiler 11 feet wide on base and 9 feet

* Reports to Navy and Treasury Departments, 1874-5.

high, three furnaces in each boiler, located between water-legs. The products of combustion return through tubes within the shell. The boiler of the "Dallas," designed for low-pressure steam, is 13 feet 9 inches long. The boilers of the two other vessels were designed for high-pressure steam, and are each 12 feet long. The steam-chimney is connected to the boiler by a large tube. The boiler of the "Dallas" has 160 tubes, 3¼ inches in diameter and 9 feet 3 inches long. The boilers of the two other vessels have each 158 tubes, 3¼ inches in diameter and 9 feet 8 inches long.

The "Rush" is propelled by a compound engine, with vertical cylinders and intermediate receiver, the pistons being separately connected to cranks at right angles. The cylinders are steam-jacketed, felted, and lagged, and are, respectively, 24 and 38 inches in diameter, with 27 inches stroke of piston. The steam is distributed to the high-pressure cylinder by a short slide-valve with cut-off plates sliding on the back. The distribution of steam to the low-pressure cylinder is effected by means of a double-ported slide-valve with lap proportioned to cut off the steam at about half-stroke. The surface condenser contains 900 square feet of condensing-surface. The air-pump is operated from the cross-head of the low-pressure engine. The circulating-pump is of the centrifugal type, operated by a small engine, directly connected. The screw is 8 feet 9 inches in diameter, with mean pitch of 14½ feet. The engine was intended to be operated regularly with a steam-pressure of 80 pounds, but during the trials, hereafter referred to, it was reduced to correspond to the pressure carried on the trial of the "Dexter."

The "Dexter" is of the inverted type, with a single cylinder 26 inches in diameter and 36 inches stroke of piston. The cylinder is not jacketed. Steam is distributed by a short slide-valve, with adjustable cut-off plates. The condenser is located outside the frame, but it and the pumps are duplicates of those in the "Rush." The engine and boiler are designed to be operated with a maximum steam-pressure of 70 pounds.

The "Dallas" is of the inverted type, with a single cylinder

36 inches in diameter, with 30 inches stroke of piston. The cylinder is not steam-jacketed, but is covered with non-conducting composition and lagged. Steam is distributed by a short slide-valve, with adjustable cut-off plates. The surface condenser is located under starboard frames, and has the same condensing surface as those in the other vessels. The air- and circulating pumps are also substantially the same. The engine and boiler are designed to be operated with a maximum steam-pressure of 40 pounds.

The experiments were made with the vessel secured to the wharf.

The coal was broken on the wharf to proper size (the vessel's bunkers having been closed and sealed) and filled into bags to a certain weight

The bags were sent on board, as ordered by the engineer on watch, he making record of the number of bags and the time of receipt; a similar record being made by one of the men on the wharf. At the end of the hour the number of bags on the fire was reported and entered in the appropriate column. The ashes were measured into buckets, of which the mean weight was ascertained and tallied as they were hoisted out.

The feed-water was measured before its return to the boiler; for this purpose a tank of boiler-plate was constructed, having a plate dividing it vertically into two equal parts. In the upper edge of the plate was cut a notch eight inches long, by which the height to which each half of the tank could be filled was determined. The mean of the weight of water in the half-tanks was 1129⅛ pounds, at a temperature of 72° Fahrenheit. In the computations for each experiment, the weight of water is reduced to correspond with mean temperature.

One of the feed-pumps was disconnected from the check-feed valve, and its discharge-pipe led to a receiving-tank placed over the two halves of the measuring-tank, into which this pump forced the water from the hot-well. The receiving-tank had two cocks, one over each half-tank, so that either could be filled from it at will.

The other feed-pump had its suction-pipe detached from the hot-well and connected with the bottoms of the two half-tanks, through a cock on each, so that the contents of either could be drawn out and discharged.

The method of measuring the water was as follows: One side having been filled, the cock above it on the receiving-tank was closed and the other over the empty half opened. When the water in the full one had settled to the edge of the notch, its cock in the feed-pipe was opened and the contents pumped into the boiler (care being taken to empty one in less time than it required to fill the other); when empty, its feed-cock was closed. When the water in the tank reached within a few inches of the notch, a gong in the engine-room was sounded to call attention, and when it reached the notch the gong was struck twice; at this instant the assistant engineer in the engine-room noted the reading of the counter, and an attendant in the fire-room noted and reported the height of water in the glass gauge on the boiler, as shown by a scale secured to it. The attendant at the tank also noted the time of filling, and the temperature when half emptied.

By this system of checks all errors of record could be detected, and it was possible to preserve and utilize any continuous run which came to an end through derangement of the engine. All parts of the tanks, pipes, and cocks were plainly visible to the eye, and had any leaks occurred therein they must have been detected.

A number of indicators were tested with steam before the trials, and a pair selected correct by a standard gauge. Indicator-diagrams were taken every twenty minutes, and the data for the columns of the log, except coal and ashes, every half hour.

It was ascertained that the pistons were tight by removing the cylinder-covers and letting on full steam-pressure.

During the first and principal experiments with each vessel the boilers were worked at their maximum power with natural draught at the dock, the fires cleaned regularly, and the cut-offs adjusted to carry a steam-pressure of about 70 pounds during

trial of the "Rush" and "Dexter," and about 35 pounds during that of the "Dallas."

At the conclusion of the principal experiments on each vessel, shorter experiments were made to determine the effect of varying the degree of expansion at the approximate steam-pressures of 70 and 40 pounds. In the case of the "Dexter" the cut-off was shortened for one experiment as much as the gear provided would permit; and for this vessel, as well as for the "Dallas," the cut-off was gradually lengthened, during other experiments, as far as the boiler would supply steam.

The long runs having demonstrated the evaporative qualities of the boilers, record was made, during the short runs, of the water used only. While these runs were in progress an officer was stationed at the tanks and one in the fire-room, in addition to the usual number on watch, to avoid the possibility of error.

The data obtained from these engines have been carefully classified and arranged by Prof. Cotterill.* They will be given presently.

Another good illustration of a marine-engine trial is the following, the report quoted being that of Sir F. J. Bramwell on the "Anthracite." As full an abstract is given as is needed to bring out the most salient parts and essential methods. In studying these results, it may be borne in mind that the efficiency of an ideal engine of similar working conditions, but free from wastes other than thermodynamic, would be about 0.26, and the weight of steam and fuel required, allowing an evaporation of 9 to 1, would be about 8 pounds and 0.9 pounds respectively.† The difference, 9 pounds of feed-water and 0.9 pounds of fuel, between the ideal and the real engine measures the wastes of the latter.

The engines are of the "direct-acting inverted" type, with surface condensation. Two cylinders are used; the after cylinder has two diameters of bore; the upper (the smaller one) is

* Cotterill's Steam-engine, p. 292 *et seq.*

† For details of computation see Rankine's Steam-engine, pp. 396–406, 410–411.

the high-pressure, and receives the steam from the boiler during the first half of the down stroke; the lower (the larger diameter) is the medium, or "intermediate," and is supplied at the up stroke with the steam which in the high-pressure did the work of the preceding down stroke. The exhaust from the bottom of the after cylinder passes into a chamber, from which is afforded the supply to the low-pressure (the forward) cylinder. Thus there is obtained in two cylinders an expansion of thirty-two times.

The surface condenser is composed of a number of close-topped, galvanized, wrought-iron tubes, standing vertically from a tube-plate, and having within them smaller tubes open at both ends, and proceeding upward from a lower tube-plate, so that the water from the sea passes up through the central tubes and down the annular spaces to the inlet of the circulating-pump.

The exhaust steam comes into contact with the exterior of the tubes, the condensed steam being drawn off by the air-pump, and returned to the hot-well, which surrounds the upper part of the condenser.

The space between the high-pressure piston and the upper side of the intermediate piston is in connection with the chamber which supplies the low-pressure cylinder.

The cylinders and their covers are heated by steam circulating through wrought-iron pipes cast into the thickness of the metal, and are very efficiently cleated, so as to prevent loss of heat.

The boiler is the Perkins boiler, formed of successive horizontal rows of wrought tubes (3 inches external diameter), connected at frequent intervals by vertical thimbles, the whole series being contained in a wrought-iron double casing, having the space filled in with vegetable black.

The boiler is supplied with distilled fresh water.

The drawings show the high-pressure cylinder to be 8 in. bore, the "intermediate" 16 in. bore, and the low-pressure 23 in. bore, but the cylinders were all somewhat smaller than the foregoing dimensions, the high-pressure cylinder is $7\frac{7}{8}$ in. diameter, the intermediate is $15\frac{13}{16}$ in., and the low-pressure is

$22\frac{13}{16}$ in. The stroke in both cylinders is 1 ft. 3 in. The diameter of the piston-rods (the areas of which have to be deducted from the area of the intermediate piston, and from that of the underside of the low-pressure piston) is $2\frac{1}{4}$ in.

Preparations for the trial had been made by weighing out 50 cwt. of "Nixon's navigation hand-picked lumps," into 50 one-hundredweight sacks. These were ranged on deck. The bunkers, which were full, were sealed up both above and below. A spring-balance was hung up on deck, close to the stokehold hatch, and an assistant caused each sack of coal to be re-weighed just before it was lowered for use. The sacks were afterwards separately weighed to obtain the net weight of the coal.

Fifty pounds of dry wood were served out, and two sacks of coal; and with these the fire was laid (the grate has an area of about 15 square feet).

The fire was lit	at 6.28 A.M.
Steam was up and the engines were turned round	at 7.18 A.M.
The height of water in the boiler-gauge was noted, and also the height in the hot-well; the still-cock being shut, was sealed in that position. The steam stop-valve was sealed in its wide-open position.	
The engines were started and the vessel got under weigh	at 7.20 A.M.
The throttle-valve was put into the position which the engineer knew, from experience, would cause the engines to run at about 130 revolutions per minute after they became thoroughly heated up, and the handle was sealed into this position, the link motion being in full gear ahead.	
The engines were provided with a Harding's counter, such as is used in the Navy, and there were pressure-gauges to show the boiler-pressure and the pressure in the chamber supplying the low-pressure cylinder, and the condenser was	

provided with a vacuum-gauge. Four Richards indicators were fitted, viz.: one to the high-pressure end of the after cylinder, one to the intermediate end of that cylinder, one to the top and one to the bottom of the forward, the low-pressure, cylinder.

The first set of diagrams was taken............ at 8.22 A.M.

The first reading of the counter................ at 8.30 A.M.

From this time until.......................... 5.45 P.M.
the counter was read at the hours and half-hours, and sets of diagrams were taken at the quarter past the hour and at the quarter to the hour. The time when each sack of coals was lowered into the stokehold was noted, and also the time when the stoker commenced to use the contents of each sack.

The last shovelful of the 15th sack was put on... at 5.18 P.M.
and it was decided to stop the trial as soon as the coal then in the fire was exhausted.

The engines ran on until they stopped of themselves, and indicator-diagrams were taken, first at each quarter of an hour and then at each five minutes during the time they were gradually stopping.

The quarter-hour diagrams were taken until .... 6.30 P.M.
when the engines were making 124 revolutions, and the five-minute diagrams were taken until the engines came to a stand.................. at 7.23 P.M.
or 12 hours 3 minutes after their start in the morning.

The water in the boiler was pumped to the same level in the gauge as that at which it had stood in the morning, and the height of water in the hot-well was noted.

The mean revolutions from 8.30 A.M. to 6.30 P.M., 10 hours, were 130.77 per minute, and from the first start to the same time being 11 hours 10

minutes, the mean revolutions were 130.4 per minute.

From the start at 7.20 A.M. to 6.30 P.M., 11 hours and 10 minutes, the engines developed an average gross indicated horse-power of 80.55, but from 6.30 to the time (7.23 P.M.) that the engines stopped of themselves, from the fire having burnt itself out, the power was gradually diminishing.

We estimate the 50 lbs. of wood as of about one-third the value of coal as fuel, say.................... 17 lbs.

The coal, 15 cwt.......................... 1680 "

1697 lbs.

The loss of water for the whole 12 hours was 23½ gallons.

The amount of lubrication was small, involving an expenditure of about one gallon of lard-oil, while cylinder and slide lubrications are in the Perkins system inadmissible and, with the metal used, unnecessary.

At the conclusion of the trial the assistants took away the four indicators, and the spring-balance with which the coals were weighed. All these were tested, with the result that the balance and the 100-lb. spring (used in the high-pressure cylinder-indicator) and some of the lighter springs were absolutely accurate, and that the variation in the others is too trifling to call for any allowance in calculating the mean pressures.

The mean pressures of the various diagrams were ascertained by dividing the areas of the diagrams obtained with the planimeter by the lengths of the diagrams.

The data obtained were as follows, as subsequently recomputed and arranged by the board appointed by the U. S. Navy Department to examine the vessel, the extraordinary pressures and ratios of expansion adopted attracting attention and making this work important:

Data and Results from the Experiment by Mr. Bramwell on the "Anthracite."

Economic results:

Pounds of coal consumed per hour per indicated horse-power 1.7114

Pounds of coal consumed per hour per net horse-power	1.9634
Pounds of coal consumed per hour per total horse-power	1.4291
Pounds of combustible consumed per hour per indicated horse-power	1.6259
Pounds of combustible consumed per hour per net horse-power	1.8653
Pounds of combustible consumed per hour per total horse-power	1.3577
Pounds of feed-water consumed per hour per indicated horse-power	17.8304
Pounds of feed-water consumed per hour per net horse-power	20.4560
Pounds of feed-water consumed per hour per total horse-power	14.8893
Fahrenheit units of heat consumed per hour per indicated horse-power	20,021.7027
Fahrenheit units of heat consumed per hour per net horse-power	22,969.9810
Fahrenheit units of heat consumed per hour per total horse-power	16,719.1503

Weight of steam accounted for by the indicator:

Pounds of steam present per hour in the first cylinder at the point of cutting off the steam, calculated from the pressure there	989.3756
Pounds of steam present per hour in the first cylinder at the end of the stroke of its piston, calculated from the pressure there	890.6505
Pounds of steam condensed per hour in the first cylinder to furnish the heat transmuted into the total horse-power developed in that cylinder by the expanded steam alone	45.1639
Sum of the two immediately preceding quantities	935.8144
Pounds of steam present per hour in the second	

...at the end of the stroke of its piston, ...om the pressure there.....	1,004.7334
...steam condensed per hour in the ...second cylinders to furnish the heat ...ted into the total horse-power de... in those cylinders by the expanded ...alone..	124.8752
...the two immediately preceding quanti...	1,129.6086
...s of steam present per hour in the third ...at the end of the stroke of its pis-...calculated from the mean of the pres-...there for the down-stroke and up-stroke of the piston	1,118.3780
Pounds of steam condensed per hour in the first, second, and third cylinders to furnish the heat transmuted into the total horse-power developed in those cylinders by the expanded steam alone....................	199.1154
Sum of the two immediately preceding quantities.	1,317.4934

Weight of water vaporized in the boiler from the feed temperature:

Pounds of steam evaporated per hour in the boiler on the supposition that this weight was equal to the weight of steam accounted for by the indicator at the end of the stroke of the piston of the third cylinder plus 121.9992 pounds condensed in that cylinder by other causes than the development of the power; this 121.9992 pounds is calculated from the weight of 147.2538 pounds condensed per hour in the third cylinder during the experiment made at the New York Navy-yard on the machinery of the "Anthracite," divided by the ratio 1.207, of the difference between

the temperatures of the initial steam in that cylinder on its piston and of the back-pressure steam against it at the commencement of the stroke in that experiment and in the present one. In the navy-yard experiment the temperature of the initial steam on the piston of the third cylinder was 245.76 degrees Fahrenheit, and the temperature of the minimum back pressure against the piston was 150.25 degrees Fahrenheit: difference 95.51 degrees Fahrenheit. In Mr. Bramwell's experiment the temperature of steam of the initial pressure on the piston of the third cylinder was 230.60 degrees Fahrenheit, and the temperature of the minimum back pressure against it was 151.47 degrees Fahrenheit; difference, 79°.13 Fahr. And $\frac{95.51}{79.13} = 1.207$, the ratio used above........................ 1.4394926

Difference between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator:

Difference in pounds per hour between the weight of water vaporized (1455.9126 pounds) in the boiler and the weight of steam accounted for by the indicator in the first cylinder at the point of cutting off the steam.. 450.1170

Difference in per centum of the weight of water vaporized in the boiler between that weight and the weight of steam accounted for by the indicator in the first cylinder at the point of cutting off the steam...................... 31.27

Difference in pounds per hour between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the first cylinder at the end of the stroke of its piston...................... 503.6782

Difference in per centum of the weight of water vaporized in the boiler between that weight and the weight of steam accounted for by the indicator in the first cylinder at the end of the stroke of its piston....................	34.99
Difference in pounds per hour between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the second cylinder at the end of the stroke of its piston....................	309.8840
Difference in per centum of the weight of water vaporized in the boiler between that weight and the weight of steam accounted for by the indicator in the second cylinder at the end of the stroke of its piston....................	21.53
Difference in pounds per hour between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the third cylinder at the end of the stroke of its piston....................	121.9992
Difference in per centum of the weight of water vaporized in the boiler between that weight and the weight of steam accounted for by the indicator in the third cylinder at the end of the stroke of its piston....................	8.47

The results of the trials of the "City of Fall River," which follow, exhibit the effect of varying conditions of operation as a simple and as a compound engine.* This steamer was a side-wheel freight-boat of the Old Colony Steamboat Co., plying between New York, Newport, R. I., and Fall River, Mass., having a compound vertical-beam engine, H. P. cylinder 44″×8′, L. P. cylinder 68″×12′, built by W. & A. Fletcher, North River Iron Works, New York, 1883, and so constructed that the high-pressure cylinder could be entirely disconnected, leaving a simple beam engine, having a steam-cylinder 68″×12′.

* Report on Trial of the "City of Fall River," Jour. Franklin Inst., July, 1881.

RESULTS OF EXPERIMENTS.

Trip No.	Date	Style of Engine.	Ports.	Distance, statute miles.	Draught of water.	Displacement, gross tons.	Running time.	Average pressure steam per gauge	Speed per hour.	Revolutions.	Revolutions per minute	Horse-power.	Coal, in pounds.	Coal per hour, pounds.	Coal per hour per horse-power	Water per hour, in pounds.	Feed-temperature of water	Water per hour per horse-power.	Tide, etc.
	1883				ft in.		h. m.		Stat. miles.						lbs.			lbs.	
1	May 3	Compound.	New York to Fall River.	179	10 7	1,948	10 33	68½	16 96	15,833	25 01	1,578	...	....	.	26,696	104 2	17	Fair.
2	May 4	Compound	Fall River to New York	179	10 5	1,908	11 35	69	15 45	17,711	25.5	1,615	....	3,300	2.04	27,718	101.6	17 16	Ahead.
3	May 9	Compound.	New York to Fall River.	179	10 7	1,948	11 18	70	15 84	17,392	25.65	1,611	36,800	3,527	2 0217	27,834	97	17 3	Ahead.
4	May 10.	Compound	Fall River to New York	179	10 6½	1,938	10 46	70	16 62	16,669	25.8	1,602	35,609	3,307	2.0388	27,854	97	17 22	Fair.
5	June 7	Simple	Fall River to New York	179	10 6	1,928	12 04	28	14 83	17,286	23 87	1,347	.	.	...	35,271	111	26.2	Ahead.
6	June 11	Simple	New York to Newport	160	10 7	1,948	10 26	66	15 34	15,287	24.42	1,477	43,000	4,121	2 8		.	....	Even
7	June 12.	Simple	Newport to New York.	160	10 6	1,928	10 40	25½	15	15,415	24 1	1,457	44,150	4,139	2 84	...	..	....	Fair.

In all cases the fires were well burnt down at the end of each trip, and, when the boat arrived at dock, fires were banked and coal put on to keep them alive while steam is blown off. During the day, about noon, more coal was put on the fires. An hour previous to departure fires were hauled down, and spread with fresh coal, to make steam, and ordinarily no further firing is necessary until half an hour after starting. On each trip indicator-cards were taken every half-hour. Water was measured by meters, readings taken every hour. The meters were tested by measuring water, under the same pressure as when feeding boilers, into a barrel, and weighing four cubic feet at a time; variations of such tests being from 61.4 to 61.5 lbs. per cubic feet. On June 10, the cut-off on the simple engine was shortened, making it easier to keep steam and to run with wide throttle. All water-measurements, power-calculations, and coal-measurements of trips Nos. 1, 2, 3, 4, 5 were reported to the Author by Messrs. Adger and Sague, the observers. The power and coal measurements of trips Nos. 6 and 7 were made by the W. & A. Fletcher Company.

Final Results.

Compound engine, 14 trips between New York and Fall River, May 15 to June 2. Average time, 11 hours $12\frac{6}{10}$ minutes; coal, 20.65 tons.

Simple engine, 12 trips between New York and Fall River, June 4 to 10. Average time, 11 hours $57\frac{6}{10}$ minutes; coal 27.42 tons.

Deducting 3 tons per trip for banking, spreading fires, donkey boiler, and kitchen (all of which is included in the amount of coal given), makes the actual consumption of coal per trip while the engine was running: for compound engine, 17.65 tons; and for simple engine, 24.42 tons.

The hull of this steamer was of the following dimensions:

Length on the load water-line	260 ft.
Length over all	273 "
Breadth of beam on load water-line......	42 "

Breadth of beam over guards...........	73 ft.
Depth of hold moulded................	18 "
Draught of water, light................	9 " 3 in.
Draught of water, loaded 600 tons......	12 "
Depth between-deck, from top of plank-shear to top of upper frame..........	11 "

The paddle-wheels were of the feathering variety, and of the following proportions:

Diameter outside of buckets............	25 ft. 6 in.
Number of buckets....................	12
Width of each bucket..................	40 in.
Length of each bucket.................	10 ft.
Distance from centre of wheel shaft to centre of eccentric actuating paddle-levers............................	12 in.
Length of arm on bucket from axle to lever-pin..........................	21 in.

The engine was found to have an efficiency of mechanism of about 83, and the paddles about 80 per cent.; their total efficiency being thus 66 per cent., or two-thirds. It was of the McNaught type. The boilers were tubular and contained—

Area of grate-surface in each boiler	115 sq. ft.
Area of water-heating surface in each boiler	3,345 "
Area of steam-heating surface in each boiler	205 "
Ratio of water-heating to grate surface...	29
Ratio of total heating to grate surface...	30.87
Weight of each boiler..................	51½ tons (net)
Weight of water in each boiler..........	27 " "

The figures in the table are deduced from their performance:

The thermodynamic efficiency of the steam as used in the compound engine on May 10 was computed according to the method given by Rankine.* The pressures of admission and of release and the mean back-pressure were obtained from the cards and the corresponding temperatures, densities, and latent heats arrived at by using the formulas given.†

* Steam-engine, § 284. † *Ibid.*, §§ 206, 255.

RESULTS OF BOILER-TRIAL.

Position of Boiler.	For'd	Aft.	For'd	Aft.	For'd	Aft.
DATE OF TEST.	May 4	May 4	May 9	May 9	May 10	May 10
Length of test (hours)	13	13	12.5	12.5	12.5	12.5
Total coal burned (lbs.)	18922	21314	18294	22083	18186	21650
Total refuse, ash, etc. (lbs.)	3699	3427	3518	3206	3449	3503
Total combustible (lbs.)	15223	17887	14776	18877	14737	18147
Percentage of refuse, ash, etc.	19.55	16.08	19.23	14.52	18.97	16.18
Total water evaporated (lbs.)	154106	183785	151057	182019	144303	176576
Average steam gauge pressure	68.5	68.5	70	70	70	70
Average height of barometer	30.70	30.70	30.70	30.70	30.5	30.5
Average temperature of feed-water (Fhr.)	102°.3	101°	97°.1	97°	67°	97°.3
Average temperature of atmosphere (Fhr.)	74°	74°	78°	78°	77°	77°
Average temperature of chimney gases (Fhr.)	435°	485°	485°	495°	493°	416°
Number of pounds of coal per hour per sq. ft. of grate	12.657	14.257	12.726	15.292	12.646	15.069
Number of pounds of water evaporated from the temperature of feed per lb. of coal	8.14	8.62	8.257	8.24	7.939	8.156
Number of pounds of water evaporated from the temperature of feed per lb. of combustible	10.117	10.27	10.22	9.639	9.797	9.732
Number of pounds of water evaporated from and at 212° F. per lb. of combustible	11.59	11.77	11.75	11.08	11.266	11.192

The following are the data and results:

p_1 = absolute pressure of admission = 11808 lbs. per. sq. ft.
p_2 = absolute pressure of release = 1363.68 " "
p_3 = mean absolute back-pressure = 704.16 " "
t_4 = absolute temperature of feed-water = 558°.36 Fahr.

The corresponding temperatures, densities, and latent heats are designated by the same subscripts.

$t_1 = 774°.50$ Fahr. $\qquad t_2 = 652°.32$
$L_1 = 131841.14 \qquad L_2 = 19000.39$
$D_1 = .1909 \qquad D_2 = .02606.$

From these data the following results were arrived at by considering the cylinders as non-conducting and the engine perfect:*

The ratio of expansion $r = 6.7167$.

Energy per cubic foot of steam admitted $UD_1 = 27183.43$ foot-lbs.

* Steam-engine, pp. 388, 389.

Heat expended per cubic foot of steam admitted H_1D_1 $= 163716.507$ foot-lbs.

Mean effective pressure, or energy per cubic foot swept through by piston,

$$\frac{UD_1}{r} = 4047.5 \text{ lbs. per sq. ft.}$$

Heat expended per cubic foot swept through by the piston, $\frac{H_1D_1}{r} = 24377$ lbs. on square foot = pressure equivalent to heat expended.

$$\text{Efficiency of steam} = \frac{UD_1}{H_1D_1} = \frac{U}{H_1} = .166.$$

Net feed-water per cubic foot swept through by piston

$$= \frac{D_1}{r} = .0284.$$

Cubic feet to be swept through by piston for each indicated horse-power per hour $= \dfrac{1980000}{M.E.P._1 = 4047.5} = 489.2$ cubic feet.

Feed-water per I. H. P. per hour $= 489.2 \times .0284 = 13.89$ lbs.

Actual feed-water $= 17.00$ lbs., nearly.

	13.89
Difference,	3.11 lbs. = 22 per cent.

due to cylinder condensation and leakage waste and other wastes not taken into account. This is an exceptionally excellent result. These wastes are usually much greater.

101. Results and Deductions and Graphical Records of these trials are rich in matters of interest to the engineer. The data obtained in the trials of the engines designed by Mr. Emery are as follows: *

* Cotterill, pp. 294–296.

DISTRIBUTION OF HEAT.

	Description of Engine.	Principal Particulars.			Useful Heat Expended per cent of Total			Losses of Heat per cent of Total Heat					Total Expenditure of Heat and Steam.	
		Boiler Pressure.	Speed of Piston	Ratio of Expansion	Useful Work	Loss in a Perfect Engine	Useful Heat	Exhaust Waste	Incomplete Expansion.	Heating Feed	Excess Back Pressure.	Other Losses	Lbs Steam per I. H. P per Hour.	Thermal Units per I. H. P. per 1'.
	"BACH."													
1.	Compound engine, low-pressure cylinder, jacketed.	95¼	150	11½	6 7	20.6	20.3	....	....	....	....	....	35.1	633
2.		94½	180	7¾	8 0	24.4	32.4	37.6	11.2	7 9	2.6	8.3	29.6	534
3.	Operated as a simple engine without jacket.	92¾	188	5¼	8.9	27.1	36.0	27.2	18.0	7.9	2.4	8 5	26½	480
4.	Operated as a simple engine, with jacket in use.	95¼	160	12¼	8.6	24 4	33 0	32 0	12.8	8.0	1.8	12 4	27.1	493
5.		95¾	185	8¼	9 7	27.5	37 2	27 9	13.4	8.1	1.5	11.9	24.1	441
6.		94	215	5	10.2	29.2	39 4	21 9	21.2	8 2	2.1	7 2	23.1	419
7.		45¼	181	2½	7.1	30 6	37.7	24 4	22 9	6.4	3.3	5.3	34.0	602
8.	Operated as a compound engine, with jacket in use.	96½	155	17	9.2	24.3	33 5	15 4	15.6	7.6	4 0	23.9	25 1	465
9		95½	193	9½	11.0	28.2	39 2	13 5	18.5	8.0	2 5	18 3	20.7	389
10		95	213	7	11.3	28.8	40 1	16 4	19 1	8.2	2.5	13 7	20.3	378
11.		95	225	5¾	11.2	28.4	39 6	15.6	19.6	8.2	2.5	14.5	20.4	382
12.		94	242	4½	11.1	30.8	41.9	15.0	19 9	8.0	1.9	13.3	21.2	385
	"DALLAS."													
13.	Simple engine, not jacketed.	50	243	5	9 4	33.1	42 5	20 4	18.1	6.8	3.2	9.0	26.7	455
14.		50	285	3½	9 4	33 1	42 5	16 3	20 8	6.8	3.6	10 0	27.0	455
15		46½	308	3½	9 4	35 8	45.2	17.8	22.1	6 2	4.3	4.4	26.9	455
16.		48½	322	2¾	8.9	33 5	42 4	19 6	21.9	6.7	3.8	5.7	28.9	480
17.		42	317	2¾	8.3	33.0	41.3	19.4	23.9	6.4	4.0	5.0	31.0	515

DISTRIBUTION OF HEAT.—*Continued.*

No.	Description of Engine.	Principal Particulars: Boiler Pressure.	Speed of Piston	Ratio of Expansion.
	"DEXTER."			
18.	Simple engine, not jacketed; leakage not included.	83½	339	4½
19.		82	366	3½
20.		81½	437	2¾
21.		55½	305	3½
22.		56¾	364	2
	"RUSH."			
23	Compound engine, both bylinders jacketed.	84	319	6½
24.		51¾	250	4
	"GALLATIN."			
25.	Simple engine, jacketed; jacket in use.	86½	255	7.3
26.		82	343	4.9
27.		60½	221	6.1
28.		58	230	5.1
29		56	246	3.7
30.		52¾	291	2.2
31.		30	206	2.0
32.		28	212	1.5
33.	Jacket not in use.	86	261	7.8
34.		83½	300	5.0
35.		59½	215	5.9

No.	Useful Heat Expended per cent. of Total: Useful Work.	Loss in a Perfect Engine	Useful Heat.
18.	10.4	29.7	40.1
19.	10.4	30.0	40.4
20.	10.3	30.6	40.9
21.	8.6	27.1	35.7
22.	7.8	26.4	34.2
23	12.7	36.3	49.0
24.	10.6	36.3	46.9
25.	11.4	32.3	43.7
26.	10.9	32.9	43.8
27.	10.2	30.1	40.3
28.	9.8	30.1	40.9
29	10.2	33.8	44.0
30.	9.0	32.8	41.8
31.	7.2	31.9	39.1
32.	6.4	30.4	36.8
33.	9.4	26.6	36.0
34.	10.6	29.7	40.3
35.	9.0	26.9	35.9

No.	Losses of Heat per cent. of Total Heat: Exhaust Waste.	Incomplete Expansion	Heating Feed	Excess Back pressure	Other Losses.	Total Expenditure of Heat and Steam: Lbs. Steam per I. H. P. per Hour.	Thermal Units per I. H. P. per 1'.
18.	21.0	19.1	8.2	3.9	7.7	22.6	411
19.	20.0	21.1	8.2	3.9	6.4	22.7	411
20.	13.7	25.2	8.0	5.2	7.0	23.0	415
21.	24.9	20.2	7.4	4.5	7.3	27.4	497
22.	26.4	21.7	7.3	4.4	6.0	30.2	548
23	....	....	....	....	....	18.4	336
24.	....	....	....	....	....	22.1	404
25.	15.2	20.1	7.8	3.1	10.1	20.5	374
26.	12.9	25.4	7.8	3.1	7.0	21.5	390
27.	13.9	22.7	7.2	5.0	10.9	22.9	420
28.	20.3	17.8	6.7	3.7	10.6	24.0	437
29	18.3	25.7	7.1	3.9	1.0	23.2	420
30.	13.8	27.1	6.7	3.5	7.1	26.5	476
31.	16.9	26.6	5.8	4.3	7.3	33.3	591
32.	17.7	25.6	5.4	4.7	9.8	37.4	664
33.	22.8	19.5	8.5	3.4	9.8	25.0	454
34.	17.8	24.4	8.3	2.0	7.2	21.9	401
35.	24.5	14.7	7.7	3.0	14.2	26.0	476

DISTRIBUTION OF HEAT — *Continued*

Description of Engine.	Principal Particulars. Boiler Pressure	Speed of Piston	Ratio of Expansion	Useful Heat Expended per cent of Total. Useful Work	Loss in a Perfect Engine	Useful Heat	Losses of Heat per cent. of Total Heat. Exhaust Waste	Incomplete Expansion	Heating Feed	Excess Back Pressure	Other Losses	Total Expenditure of Heat and Steam. Lbs Steam per I. H P per Hour.	Thermal Units per I H P per 1'.
"GALLATIN."—(*Cont'd.*)													
36. Jacket not in use.	58	254	3.7	9.8	30.3	40.1	17.7	24.9	7 5	2.0	7 8	24 0	437
37.	52½	278	2.2	8 6	35.4	44.0	16.0	22.0	6 3	2 0	9 7	28.1	596
38.	29½	200	2.0	5.9	25.8	31.7	31.5	21.9	5.9	4.1	4 9	40 4	720
39.	27½	204	1.5	5.4	25.8	31.2	28 7	24.8	5.4	4.0	5 9	44.2	787
40 Jacket supplied with steam of 85 lbs. Difference of temperature 68°.	28½	205	1.8	6.9	27.5	34.4	15.0	31.9	5 8	4.1	8.8	34.1	616
41.	29	211	1.5	7.1	29.5	36 6	13.6	33.7	5.7	4.4	6 0	34 9	605
42 Without vacuum; jacket in use.	84½	247	4 1	9.9	64.1	74.0	4.0	1.8	4 7	....	15.5	25.9	431
43.	82	266	3.5	9.4	61.7	71.1	6 7	2.7	4.5	....	15.0	27.3	456
44. Without vacuum; jacket not in use.	85½	233	4.4	8.6	55.4	64 0	13 4	1.5	4.9	....	16.2	30.0	499
45	81	258	3.5	8.8	58.1	66 9	10.1	3.8	4.5	....	14.7	29.4	488

The expenditure of heat in the last column is reckoned from the temperature of the feed-water. The number obtained by dividing 42.75 by the expenditure of heat thus reckoned is the true efficiency of the engine. Experiments 42–45 are exceptions, made without vacuum. In calculating the expenditure of heat, the boiler has been supposed to supply dry steam; the results obtained are too large, though not much too large, as there is no reason to believe the amount of priming considerable.

The columns headed "Useful Heat Expended" show the useful work done, together with the corresponding *necessary* loss, expressed as a percentage of the total heat expended. The first of these columns is the absolute efficiency of the engine, and the third the efficiency relatively to a *perfect* engine working between the same limits of temperature.

The five remaining columns show heat unnecessarily lost—

(1) By the exhaust waste, by transmission of heat to the exhaust steam, and by external radiation less the heat given out by piston friction and the effects of compression.

(2) By incomplete expansion, by the amount of work which the steam discharged from the cylinder might do by expanding down to the pressure of the condenser.

(3) By misapplication of heat in heating the feed, raising the temperature of the water by direct heat instead of by compression of the exhaust steam.

(4) By excess back-pressure, the difference between the actual back-pressure and the pressure corresponding to the temperature of the condenser.

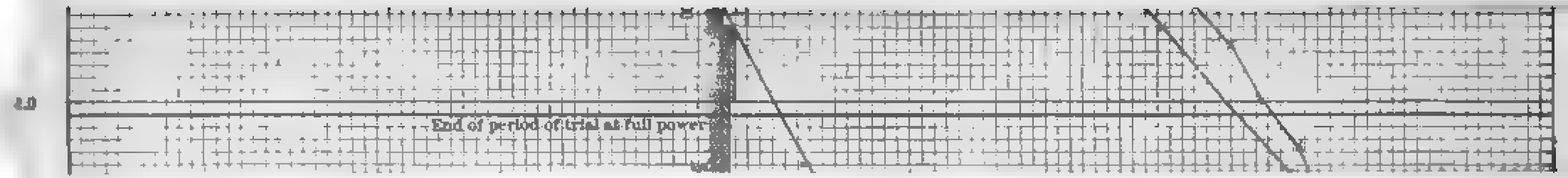
(5) By other losses, by clearance and wire-drawing and by misapplication of heat during expansion.

All the results are given as percentages of the total expenditure; but by multiplying by that expenditure, and dividing by 100, they may be expressed in thermal units per I. H. P. per minute; or, by multiplying by the consumption of steam and dividing by 100, they may be expressed in pounds of steam per I. H. P. per hour.

These computations have been made very carefully by Pro-

fessor Cotterill, and interesting conclusions are reached by their study: The large losses by exhaust waste, 27 per cent., when in the first set of these trials the steam-jacket was dispensed with, and the exaggeration of that loss by increased ratios of expansion; the reduction of this loss in the next set of four trials, by the use of the jacket; the increase of waste invariably, with increase of surface exposed; the great gain in this direction by compounding, as seen in the final set of five "Bache" trials; while the reduction of exhaust waste is partially compensated by increased liquefaction in the high-pressure cylinder. The net final advantage of compounding all these effects and variations of condition are well shown by these data. The work may be studied in detail in the treatise from which the figures are quoted. The trials of the "Dexter" developed the gain by increased piston-speed which might have been anticipated; and those of the "Rush" exhibited, again, the gain due to compounding.

The cylinders of the compound engine of the "Rush" were steam-jacketed, and those of the non-compound engines were not steam-jacketed. A new non-compound engine with steam-jacketed cylinder and a boiler designed for high-pressure steam having been completed for the revenue-steamer "Gallatin," a series of trials of the machinery of that vessel, with and without steam-jacket in use, for comparison with the trials detailed in the report, were made, and the data are similarly arranged for comparison in Nos. 25 to 45 of the table. The first twelve or fifteen trials illustrate again the good effect of steam-jacketing in reducing wastes and permitting a great increase in the best ratios of expansion at any one pressure in the compound engine. The "Gallatin" trials show similar effects in the non-compound engine. This latter engine consisted of a pair of single cylinders 34 inches in diameter and 30 inches stroke; and the trials were conducted substantially as were those of the other ships. Examining the table, it is seen that all the earlier of this set of trials exhibit in a very decided manner the good effect of the jackets, reducing waste by cylinder condensation, and increasing the ratios of expansion at maximum efficiency,


End of period of trial at full power

and also the gain due to increasing pressures. A gain is observed in the experiments in which the jacket is filled with steam of pressure exceeding that within the cylinder, but it is not great. The final set of three trials as a non-condensing engine, in effect, and both with and without jacket, exhibit considerable reduction of cylinder condensation; while the good work of the steam-jacket is very observable.

The work of the engineers on the trial of the "Anthracite," as given in the preceding article, having been challenged, a second trial was made, March 19th, 1881, obtaining very similar results, the coal amounting to 1.795 pounds per indicated horse-power per hour, including the fuel used in starting fires. These results are beautifully exhibited in the accompanying chart (Fig 124).

Graphical Records, such as these, are peculiarly valuable, both as exhibiting the progress of the trial and as a check upon the written log. As is seen in the chart, the several curves, each made to an exact scale laid down on the margin, show the principal data, in their absolute values at any given time, and also their variations with the progress of the trial. Comparing this set of curves with the figures given in the log and tabulated in the preceding article, it is easy to trace out any desired set of relations and to check the one by the other. The accuracy of the observations is also to a certain extent indicated by the smoothness of the line. Any seriously incorrect value would be immediately detected by its location outside the general trend of the curve of which it should locate a point. The lines showing revolutions, work, and coal-consumption are almost perfectly straight from end to end; those showing speed and power exhibit more strikingly the slowing down at the end of the trial. The whole is a good model of this kind of record.

Another set of illustrations of the beauty, convenience, and value of graphical records is found in those representing the results of trials of the Italian ironclad "Lepanto," which follow (Fig. 125). The "Lepanto" is a steel ship of upwards of 18000

I. H. P. of 400 feet length, 72¾ feet beam, 28½ feet draught, 46 feet depth, and of about 14000 tons displacement.*

The vessel has eight "drum" marine boilers and sixteen of the locomotive type; and blast is supplied by twenty fans to the 24 furnaces of the one set and the 32 furnaces of the other. There are four sets of compound engines. The following are the principal data:

	Marine.	Locomotive.
Grate area in boilers, ...	478.4 sq. ft.	675.2 sq. ft.
Heating-surface, ...	15,360 "	26,720 "
Ratio, ...	32.1	39.6
3 steam-cylinders (each engine), diam., ...		54 in.
Stroke of pistons, ...		39
Revolutions per min., ...		96
Maximum I. H. P., ...		18,000
Condenser surface, ...		31,300 sq. ft.
2 series of diam., ...		20½ ft.
Blades, in number, ...		3
Pitch, ...		20½ ft.
Surface of blade, each screw, ...		80 sq. ft.

The condensed and tabulated data of two of these trials are given in the table opposite.

These trials were made in the Gulf of Genoa, under the usual external conditions of work at sea. All the engines and boilers worked well. The blast was introduced into the fire-rooms, which were sealed to retain the air, under pressures varying up to 2½ and 3½ inches of water; while the consumption of fuel was from 45 to 70 pounds per square foot of grate per hour. The curves, Fig. 125, exhibit the performance of this great ship most admirably.

The E. H. P. (effective horse-power) curve, *aa*, corresponds to a displacement of 14,784 tons, the mean displacement of the "Lepanto" at the various trials.

bb is the I. H. P. curve.

dd is the "indicated thrust curve," from the I. H. P. curve.

* Major Soliam, Trans. Inst. Naval Archts., 1888.

STEAM TRIALS OF THE ITALIAN IRONCLAD "LEPANTO."

	Trials. A.	Trials. B.
Sea	Calm	Rather rough
Wind	Light N. O.	Fresh N. O.
Mean draught ... ft.	30′ 4″	30′ 3″
Area of midship section ... sq. ft.	1,999	1,993
Displacement ... tons	14,860	14,810
Wetted surface ... sq. ft.	36,500	36,430
Mean speed of ship ... knots	7.25	13.3
Indicated horse-power	1,004	5,714
Number of boilers used	2 Oval	8 Oval
Number of engines used	2	4
Mode of action of engines	Compound	Direct
Area of fire-grate used ... sq. ft.	94.1	478.4
Heating surface used — Tubes	3,488	13,952
Heating surface used — Total	3,840	15,360
Mean steam-pressure in lbs. per sq. in. — Oval boilers	50	37
Mean steam-pressure in lbs. per sq. in. — Locom. "	....	....
Mean steam-pressure in lbs. per sq. in. — Engine-room	48	34
Mean air-pressure in inches of water. — Oval boiler stokeholes	Natural	0″.94
Mean air-pressure in inches of water. — Locom. " "	....	....
Cut-off — In H. P. cylinders	0.1	0.1
Cut-off — In L. P. "	0.6	....
Ratio of expansion	11.1	5.56
Mean pressure in lbs. per sq. in. — H. P. cylinders	15.3	15.4
Mean pressure in lbs. per sq. in. — L. P. "	6.8	....
Mean vacuum in condensers ... ins.	28.6	28.6
Revolutions per minute	38.8	68.73
Apparent mean slip ... per cent.	6.4	4.6
Mean speed of piston per minute ... ft.	252.2	445.25
I. H. P. per sq. ft. of grate	10.7	11.9
Heating surface per I. H. P. in sq. ft. — Tubes	3.48	2.45
Heating surface per I. H. P. in sq. ft. — Total	3.82	2.69
Coal used per hour, in tons	0.9	6.9
Coal used per I. H. P. per hour, in lbs.	2.02	2.75
Coal burnt per sq. ft. of grate per hour. — Oval boiler lbs.	21.3	32.3
Coal burnt per sq. ft. of grate per hour. — Locom. " "	....	....
Steam used per I. H. P. per hour as shown by indicator-cards	16.1	18.2

The dotted line at the bottom of curve *dd* shows the increase of thrust due to the friction of the forward engines, acting at low power with the after engines. According to this the initial load friction of the engine would be about 7.5 per cent. of the load at full power.

ff is the "curve of the net resistance of the ship," from the E. H. P. curve.

The undulation characteristic of the E. H. P. curve, *aa*, and of the net-resistance curve, *ff*, at about 16.5 knots, is reproduced on the I. H. P. curve, *bb*, and on the indicated-thrust curve, *dd*.

Curve *cc* gives the ratio $\frac{\text{E. H. P.}}{\text{I. H. P.}} = p$, viz., the propulsive

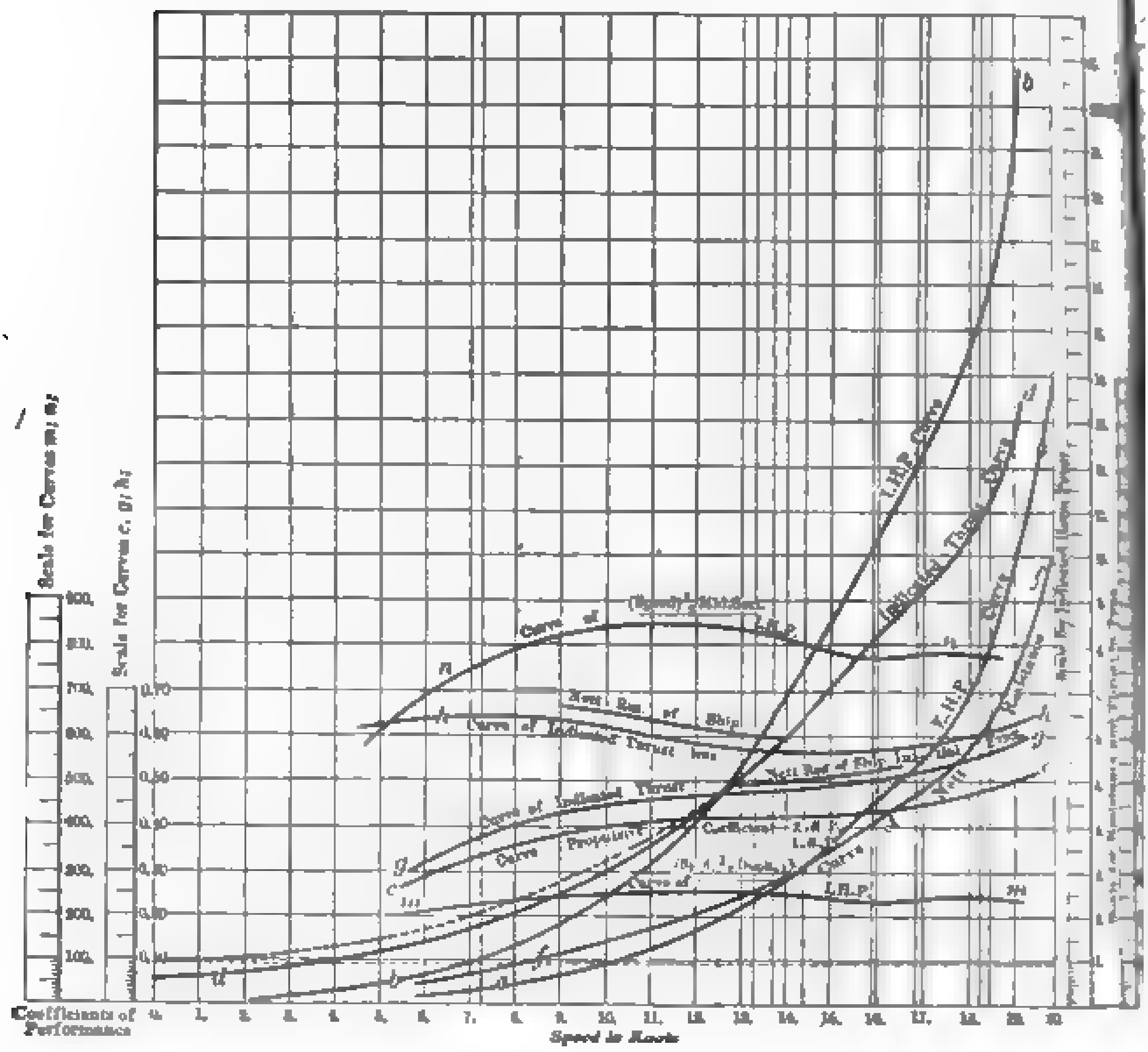


FIG. 125*a*.—CURVES OF SHIP PERFORMANCE, "LEPANTO."

coefficient or the "net total efficiency of propulsion," which slightly increases at the higher speeds when it approaches to the standard value 0.50.

Curves *mm* and *nn* give the "coefficient of performance" for displacement and midship section.

Curve *gg* gives the ratio between the net resistance of the ship and the indicated thrust.

Curve *hh* gives a similar ratio when the initial friction of the engines is taken from the indicated thrust.

Curve *rr* in Fig. 3 gives the I. H. P. in function of revolutions.

Some of the most remarkable work done by marine engines has been seen in the performance of torpedo-boats, at high

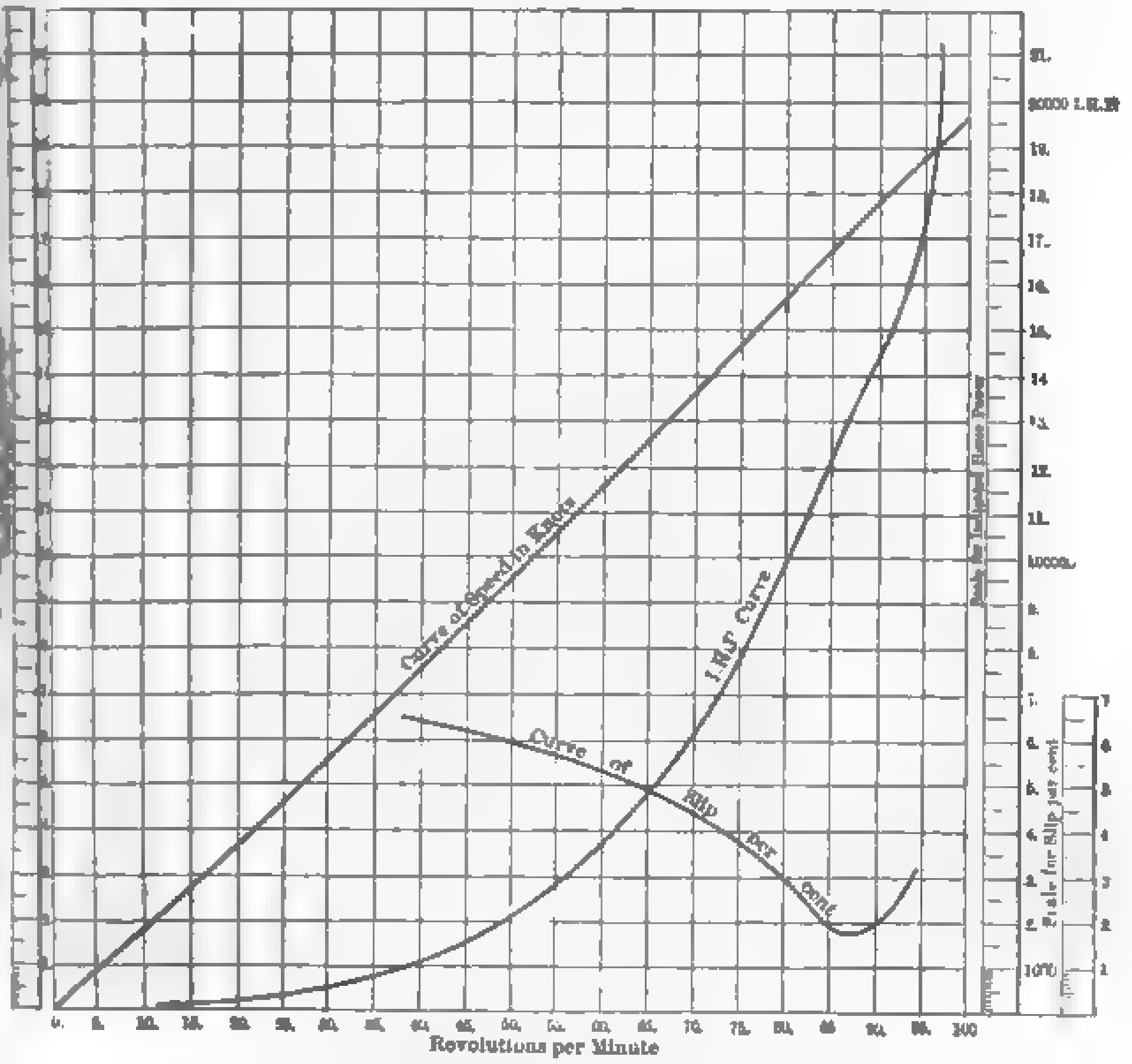


FIG. 129b.—CURVES OF SPEED AND POWER, "LEPANTO."

speed, with high steam-pressure, forced draught, and high evaporation.* Thus, in the case of a Thorneycroft boat in which the water-tubular boiler is used, with engines developing 89 indicated horse-power, the evaporation duty was 13.4 (?) of water per pound of coal, and the fuel-consumption was

* See Engineering, Feb. 22, 1889, p. 117

2.22 lbs. of coal per indicated horse-power per hour. In another experiment, with an air-pressure of 0.27 in. and the engines developing 282 indicated horse-power, the evaporative duty was 12.48 lbs. of water per pound of coal, and the coal-consumption 1.98 lb. of coal per indicated horse-power per hour. On another occasion, with an air-pressure of 0.49 in. and the engines developing 449 indicated horse-power, the evaporative duty was 12 lbs. of water per pound of coal, and the coal-consumption 1.99 lb. per indicated horse-power per hour. On a fourth trial, with 2 in. of air-pressure and the engines working with one boiler, namely, 775 indicated horse-power, the evaporative duty was found to be 10.29 lbs. of water per pound of coal, and the coal-consumption 2.26 lbs. of coal per indicated horse-power per hour.

On the natural-draught trials the temperature of the chimney gases was 421 deg. Fahr.; and on the full-speed trial, when the boiler was supplying steam for 775 indicated horse-power, the temperature was 777 deg. Fahr.

The evaporative value of the coal was found by calculation from its chemical constituents to be equal to 15.41 lbs. of water per pound of coal from and at 212 deg. Fahr. The efficiency of the boiler on the natural-draught trial was therefore 87 per cent. The efficiencies of nearly 78 per cent. with an air-pressure of 0.49 in., and over 60 per cent. with a 2-in. air-pressure, are also reported. The priming is uncertain, however.

The weight of the boiler, with all its fittings and mountings and water, was 9.84 tons, 77.8 indicated horse-power per ton of boiler at full power—a result which compares favorably with that obtained with the locomotive boiler.

102. Pumping-engine Trials offer, perhaps, the best opportunities to secure thoroughly complete records, and to effect satisfactory solutions of the problems arising in this branch of engineering. The following, which are presented purely as illustrative examples, are good illustrations both of method and result:

A trial of a small pumping-engine built by the late George H. Corliss and supplied to the city of Providence, R. I., in 1882

was made. The results were exceptionally good and the report is concise and a model in its way. It is as follows, and is submitted by the city engineer :*

Three days previous to the trial, a test was made to ascertain the actual quantity of water delivered into Sockanosset Reservoir, which is situated about one mile from the engine-house, as compared with the theoretical displacement of the plungers of the pump. This was determined by means of a weir located at the gate-chamber of the reservoir. Observations were taken at this weir in a thorough and careful manner while the engine was pumping at the rate of 9065000 gallons per 24 hours; and it was found that, for every 100 gallons displaced by the plungers of the pumps, $99\frac{8}{100}$ gallons were delivered into the reservoir. This is the most favorable result that has been brought to our knowledge. At different times, May 19th and 20th, the engine pumped at the rate of upwards of 12000000 per 24 hours.

The trial was commenced Monday morning, May 22d, and ceased Saturday evening, May 27th.

At 6.50 A.M., May 22d, clean fires were started under the three boilers used during the trial. There was no steam up at the time, as the fires had been allowed to run down on the previous Saturday afternoon. During the trial, the fires were banked each night at the end of the day's run, with the exception of the last day, when they were run down, as was done the previous Saturday.

The average time run per day for the six days was 12 hours 27 minutes and 30 seconds. The coal used was "Cumberland," or bituminous, coal, and the wood used was estimated to be equal to 40 per cent. of its weight in coal. In calculating the duty, no allowance was made for the ashes, clinkers, etc., taken from the grates and ash-pits during the trial or at its completion. During the trial the engine pumped 28360162 gallons, or at the rate of 9105604 gallons per 24 hours, and developed a duty of 113271000 foot-pounds for every 100 pounds of coal, considering the total amount of coal consumed, or up-

* Report of Chief Engineer, City of Providence, 1882.

wards of 13 per cent. in excess of the duty guaranteed in the contract; or if the coal used in kindling the fires and banking is deducted from the total coal consumed during the trial, leaving only the amount used while the engine was in operation, the duty on that amount would be 138035000 foot-pounds per 100 pounds of coal. . . .

During all the tests that had been made, the machinery had worked with ease, without interruption from any cause, and so smoothly as to inspire entire confidence in its reliability.

The table opposite shows the daily and aggregate duty, and a summary of the principal records of the trial:

The following are the essential facts and data of the trial of a pumping-engine designed by Mr. H. F. Gaskill for the city of Saratoga Springs, N. Y., as reported by Mr. Chas. T. Porter in June, 1883, and are abstracted from his report of July, 1883. The boilers were of the cylindrical fire-tube type, the engines compound. The methods of test were substantially those described as standard. A steam-jet gave a mild forced draught.

Dimensions and Performance of Boilers.

Type of boiler, cylindrical tubular.	
Number of boilers	2
Diameter of boilers........................	66 in.
Length of boilers........................	18 ft.
Number of tubes in each boiler..............	87
External diameter of tubes.................	3 in.
Arrangement of tubes: horizontal and vertical rows.	
Least distance between tubes and shell of boiler	4.5 in.
Height of crown of boiler above tubes	24 in.
Heating surface in both boilers, as follows:	
One half the shells, 311; tubes, 2,460. Total	2,771 sq. ft.
Furnace, width..........................	6 ft.
" depth	5.75 ft.
" length of opening between grate-bars	5 ft.
" " " these openings uncovered..	4.25 ft.

DAILY AND AGGREGATE DUTY AND A SUMMARY OF THE PRINCIPAL RECORDS.

Date, 1882.	Hours run.	Fuel Consumed. Coal used in pounds. Starting Fires.	Running.	Banking.	Total.	Wood in pounds.	Per cent. of Ashes and Clinkers.	Head in feet.	Gauges in Engine-room. Steam in pounds.	Vacuum in inches.	Receiver in pounds.	Gallons pumped.	Duty per 100 pounds of Coal. Calculated on Coal used in running.	Calculated on total Coal fed into furnace.
	hrs. min.													
May 22d.......	12 21	1,238	5,023	....	6,261	450	5.8	174.50	107	26.5	8.0	4,696,321	136,058,000	106,104,000
" 23d.......	10 25	569	5,000	411	5,8[illegible]	...	8.0	174.85	110	26.2	8.6	4,712,580	140,155,000	116,499,000
" 24th.......	10 8	608	4,878	425	5,911	...	7.7	174.78	107	26.3	9.3	4,627,230	138,191,000	114,041,000
" 25th.......	10 41	620	5,000	418	6,038	..	8.4	175.12	109	26.3	9.2	4,771,699	139,303,000	115,356,000
" 26th	10 30	612	5,082	428	6,122	...	8.7	175.26	108	26.2	9.9	4,754,385	136,654,000	113,440,000
" 27th.. ...	10 40	607	5,082	420	6,109	...	10.1	175.30	109	26.1	7.4	4,707,932	137,965,000	114,771,000
Aggregate of the whole run ...	74 45	4,247	30,069	2,104	36,[illegible]	450	9.1	174.90	108	26.2	8.6	28,360,162	138,035,000	113,871,000

Average gallons raised per day, 4,726,694.

Average gallons raised per revolution, 174.89.

Furnace, width of each grate-bar	1.75 in.
" " " air-spaces	.5 "
" square feet of grate-area, under both boilers	51
" of which the proportion in air-space is only	.22
Horizontal flues, as follows:	
Three times the length of each boiler . . 54 ft.	
From wall between boilers to chimney . 24 "	
Total	78 ft.
Internal diameter of common flue and chimney	3 ft.
Height of chimney above flue	75 "
Square feet of heating surface for each square foot of grate-area	54.3
Square feet of heating surface for each indicated horse-power	14.9

Performance of Boilers, on the Test for Duty, 48 *Hours.*

Coal burned	16,860 lbs.
" " per hour, on each square foot of grate	6.9 lbs.
Water evaporated	155,961 "
" " by combustion of 1 pound of coal	9.25 lbs.
Average temperature of feed-water	71.5 deg.
" pressure of steam, by gauge	73.66 lbs.
Number of thermal units contained in 1 pound of steam	1,211.1
Equivalent evaporation from 212°, under one atmosphere	10.916 lbs.
Evaporation per hour from each square foot of heating surface	1.175 "

The engines had a detachable valve-gear; the valves between the high- and low-pressure cylinders and the exhaust-valves are gridiron slides.

All the cylinders and their heads are steam-jacketed. The water formed by condensation in the jackets is returned to the boilers.

Cylinder Dimensions.

Distance between centres of cylinders.......	45.5 in.
Length of stroke in all cylinders..........	36 in.
Number of high-pressure cylinders..........	2
Diameter of " " "	21 in.
" " " " piston-rods........	3 "
Area of each piston, average of both faces...	342.83 sq. in.
Piston displacement......................	12,342 cu. in.
Waste-room, in clearance and passages......	494 "
Proportion which this adds to piston-displacement...............................	.04
Addition to length swept through by piston.	1.44 in.
Total amount of displacement and waste-room................................	7.428 cu. ft.
Number of low-pressure cylinders..........	2
Diameter " " "	42 in.
Number of rods in each piston.............	2
Diameter " " "	3.5 in.
Area of each piston, average of both faces...	1,375.8 sq. in.
Piston-displacement......................	49,528.8 cu. in.
Waste-room in clearance and passages......	1,347 "
Proportion which this adds to piston-displacement..............................	.0272
Addition to length swept through by piston.	.979 in.
Total amount of displacement and waste-room................................	29.442 cu. ft.
Proportion which is added by this waste-room to the total capacity of high-pressure cylinder..............................	.105

Performance of Engines on Forty-eight-hour Test for Duty.

Average number of revolutions made per minute....	19.37
" piston speed, in feet, " " "	116.22

Pressure of the atmosphere, in pounds on the square inch	14.5
Average horse-powers exerted in high-pressure cylinders	109.2
Average horse-powers exerted in low-pressure cylinders	76.55
Average horse-powers exerted, total	185.75
Pounds of coal burned per hour	351
" " " " " horse-power per hour	1.9
" " water evaporated per hour	3,250
" " " " " horse-power per hour	17.5
" " " " " hour for each horse-power exerted in the high-pressure cylinders	30
Total average back-pressure in high-pressure cylinders, pounds	12.5
Pounds of water per hour accounted for by indicator:	
1. At point of cut-off in high-pressure cylinders	2,061
Proportion of the quantity evaporated	.634
2. At point of release in low-pressure cylinders	2,880
Proportion of the quantity evaporated	.886
Temperature of the steam in all jackets, degrees	318.7
" " high-pressure exhaust (average)	204
" " low " "	153

The boilers were large, and the high duty was obtained from them at a slow rate of combustion and of evaporation.

The coal was hand-picked and left only 4.73 per cent. of ashes and cinders. The whole quantity burned, assuming the fires to have been in the same condition at the close as at the commencement of the trial, was 20330 pounds, which shows a duty of 106838000 foot-pounds for each 100 pounds of coal consumed. The above assumption, however, was not fully warranted, since by omitting the first twelve hours the remaining forty-eight hours show a duty of 102340000 foot-pounds for each 100 pounds of coal consumed, and this may be taken

as the real duty. It exceeded the stipulated duty by nearly twenty-eight per cent.

Studying the indicator-diagrams, it was found that 0.366 of the steam was condensed in the high-pressure cylinders, and exists in the state of water at the point of cut-off.

During the expansion in the high-pressure cylinders, in addition to the re-evaporation of water formed by the conversion of heat into work, about one half was re-evaporated.

At the end of stroke in the large cylinder 88.6 per cent. of the steam from the boilers is found in the state of vapor.

As a general rule, the higher the terminal pressure in the first cylinder the greater is the fall in passing into the second cylinder. While this fall affords a measure of the heat lost at that point, it suggests what the condensation of the entering steam must be in cylinders in which, although their surfaces have just been exposed to the same cool vapor, the pressure does not fall, because the supply of steam is unlimited.

The economy attained, although excellent, is far short of that which such engines are capable of; the economy in the high-pressure cylinders, taken alone, is inferior to that obtained in first-class non-condensing engines; and further economy is to be obtained by preventing, in a greater degree, the transfer of heat from the steam in the steam-pipe to the condenser without doing work. This can be done either by employing a higher piston speed or by moderate superheating of the steam. A combination of these would, doubtless, according to Mr. Porter, effect a further increase of from twenty to twenty-five per cent. in the duty obtained.

Dimensions and Performance of Pumps.

Number of pumps	2
Aggregate length of the two chambers in each pump..........................	7 ft. 6 in.
Width of the two chambers in each pump .	2 ft. 3 in.
Height of the two chambers in each pump.	2 ft. 11 in.
Capacity of each pump-chamber, in cubic feet	25

Diameter of plunger, inches	20
" " rod	4
Area of plunger, mean of two faces, square inches	307.88
Stroke of plunger, inches	36
Displacement of plunger, in cubic feet	6.41417
" " " " gallons	48
Number of double strokes of each plunger per minute	19.37
Mean velocity of plunger, in feet, per minute	116.22
Number of valves acting together	84
Diameter of each valve-opening, inches	1.3125
Total area opened by valves, square inches	113.6
Lift of each valve, inches	.3125
Mean velocity of water through valves, in feet, per second	5.25
Weight of each valve, pounds	.5625
Weight of each valve for each square inch of opening, pounds	.416
Resistance to current on each square inch of opening, pounds	.26
Totol loss of efficiency in pounds on square inch	$(.416 + .26) \times 2 = 1.352$
Mean excess of pressure in the delivery-main, pounds on square inch	84.57
Proportion of power lost in passage through valves	.016
Time occupied by valves in closing, seconds	.04
Motion of plunger while valves are closing, inches	.07
Proportion of stroke lost in closing of both admission- and delivery-valves	.004
Net delivery from both pumps per double stroke, gallons	190
Net delivery from both pumps per double stroke, pounds	1,584

Delivery per day at eighteen revolutions per minute, gallons	4,924,800
Number of double strokes made in 48-hour test	55.779
Net foot-pounds of work done on each double stroke	309,338
Net foot-pounds of work done by consumption of 100 lbs. of coal	102,340,000

In contrast with the results of trial of more modern types of pumping-engine, the following are data obtained from test of a well-designed and well-constructed engine of the Cornish type, as given the Author by the City Engineer of Providence, R. I., the official having charge of the engine:

RECORD OF A WEEK'S RUN OF THE CORNISH ENGINE, PROVIDENCE, R. I., 1882.

DATE.	Hours run.		Number of strokes.	Strokes per minute.	Average length of stroke.	Total head against pump.	Temperature of water in pump-well. Deg. Fahr.	Weight of c.ft. of water, av. temp.	Wood used to start fires.	Coal value of wood used.
	h.	m.			Feet.	Feet.		lbs.	lbs.	lbs.
Monday, Feb. 6th......	13	22	5505	6.864	10.79	163.57	34°	62.378	866	289
Tuesday, Feb. 7th......	12	50	5458	7.088	10.88	167.43	34	62.378	....	....
Wednesday, Feb. 8th..	12	52	5466	7.080	10.91	168.83	36	62.380	....	....
Thursday, Feb. 9th....	12	56	5468	7.046	10.90	168.74	37	62.381	....	....
Friday, Feb. 10th......	13	01	5433	6.956	10.85	169.00	37	62.381	....	....
Saturday, Feb. 11th....	12	51	5459	7.080	10.95	170.23	37	62.381	....	....
Totals and averages....	77	52	32789	7.018	10.88	167.93	36°	62.380	866	289

DATE.	Coal used for banking.	Coal used starting fires and pumping.	Coal value of total fuel consumed.	Ashes.	Per cent. of ashes.	Gallons pumped.	Rate of pumping per 24 hours.	Duty calculated on total fuel consumed in foot-pounds per 100 lbs. coal.
	lbs.	lbs.	lbs.	lbs			Gallons.	
Monday, Feb. 6th......	....	8747	9036	395	$4\frac{38}{100}$	3,382,944	6,074,114	51,065,000
Tuesday, Feb. 7th......	750	7679	8429	569	$6\frac{75}{100}$	3,382,038	6,324,851	56,012,243
Wednesday, Feb. 8th..	486	7681	8167	526	$6\frac{44}{100}$	3,306,335	6,335,120	58,568,000
Thursday, Feb. 9th....	474	7818	8292	617	$7\frac{44}{100}$	3,394,463	6,200,004	57,603,853
Friday, Feb. 10th......	512	7749	8261	620	$7\frac{48}{100}$	3,357,264	6,100,000	57,274,500
Saturday, Feb. 11th....	500	7600	8153	748*	$9\frac{17}{100}$	3,404,421	6,358,452	53,276,657
Totals and averages....	2722	47274	50338	3475	$6\frac{8}{10}$	20,317,465		56,522,364

Area of plunger 8.319 sq. ft. This area is used by the Water Department.
Loss of action determined by weir, $8\frac{45}{100}$%. Duty calculated on actual quantity delivered.

* Included in this amount was found 53 lbs. unconsumed coal.

The duty of *small steam-pumps* has been found to range very low.

A vacuum-pump tested by Mr. Emery in 1871 gave a duty on the above basis of $4\frac{7}{10}$ millions; one tested by Mr. J. F. Flagg at the Cincinnati Exhibition in 1875, reduced to the same basis, gave a maximum duty of $3\frac{26}{100}$ millions.* Several vacuum and steam pumps gave duties reported as high as 10,000,000 to 11,000,000, small steam-pumps doing no better than vacuum-pumps. Experiments made at the American Institute Exhibition of 1867 showed that medium steam-pumps do not, on the average, utilize more than 50 per cent. of the indicated power in the steam-cylinders, the remainder being absorbed in the friction of the machine, but more particularly in the passage of the water through the pump.* All steam-pumps, nearly, require that the steam-cylinder shall have 3 to 4 times the area of the water-cylinder to give sufficient power when the steam is accidentally low; hence, the net, or effective, pressure forms a small percentage of the total pressure, and this, with the large extent of surface exposed internally and externally, and the total absence of expansion, makes the waste very great. A pump tested by Mr. Emery required 120 pounds weight of steam per indicated horse-power per hour, and it is believed that the cost will rarely fall below 60 pounds. As only 50 per cent. of the indicated power is utilized, ordinary steam-pumps rarely require less than 120 pounds of steam per hour for each horse-power utilized in raising water; which is equivalent to a duty of but 15,000,000 foot-pounds per 100 pounds of coal, assuming 10,000 thermal units per pound, as in the following

The following examples of pumping-engine data illustrate a very accurate system† proposed by Mr. Barrus:

This duty of engines is expressed by the following formula:

$$\text{Duty} = \frac{\text{Foot-pounds work done}}{\text{Thermal units of heat consumed}} \times 1000000 = \frac{[C\,V\,w\,N - L] \times [H + s + h]\,1000000}{\text{Thermal units of heat consumed.}}$$

* Am Machinist, Sept 1878, p 2

† Lond. Engineering, Feb. 15, 1889, p. 170.

in which

V = volume of piston-displacement, one stroke, cubic feet.

w = weight of one cubic foot of water.

N = number of strokes during trial.

H = head in feet corresponding to indication of pressure-gauge on force-main.

h = heat in feet corresponding to indication of vacuum-gauge on suction-main.

(This is a minus quantity where there is a head of water on the suction-main and pressure-gauge is used.)

s = vertical distance in feet between the centres of two gauges.

L = total leakage of plungers during trial, estimated from results of leakage test with pump at rest.

C = correction for air admitted into the pump = proportion of the stroke during which the pump is subjected to the full discharge pressure, measured from the indicator-diagram.

Thermal units of heat consumed = weight of water supplied to boiler by main feed-pump × total heat of steam of boiler-pressure above temperature of main feed-water, plus weight of water supplied by jacket-pump × total heat of steam of boiler-pressure above temperature of jacket-water, plus weight of any other water supplied by total heat above its temperature of supply. The total heat of the steam is corrected for the moisture or superheat which the steam may contain. For moisture, the correction is subtracted, and is found by multiplying the latent heat of the steam by the percentage of moisture, and dividing the product by 100. For superheat, the correction is added, and is found by multiplying the number of degrees of superheating (*i.e.*, the excess of the temperature of the steam above the normal temperature of saturated steam) by 0.48. No allowance is made for heat added to the feed-water which is derived from any source except the engine or some acces-

sory of the engine. Heat added to the water by the use of a flue heater at the boiler is not to be deducted. Should heat be abstracted from the flue by means of a reheater connected with the intermediate receiver of the engine, this heat must be included in the total quantity supplied by the boiler.

The following examples are given to illustrate the method of computation. The figures are not obtained from tests actually made, but they correspond in round numbers with those which were so obtained.

First Case.—Jacketed compound fly-wheel engine. Jacket-water returned to the boiler by gravity. Jet condenser with air-pump, operated by main engine. Feed-pump driven by main engine supplied with water from hot-well, which receives drip from intermediate receiver. No heaters.

1.	Boiler-pressure by gauge, ...	100 lbs.
2.	Capacity of pump-displacement one stroke (V), ...	12.5 cub. ft.
3.	Number of strokes during trial (N), .	140,000
4.	Pressure by gauge on force-main, .	80 lbs.
5.	Vacuum by gauge on suction-main, .	4.8 in.
6.	Vertical distance between gauges (s),	10 ft.
7.	Temperature of water in pump-well,	60 deg.
8.	Leakage of pump determined by trial at rest (L), ...	546,000 lbs.
9.	Kind of pump-diagram, .	Rectangular (or $C =$ unity)
10.	Weight of water supplied by feed-pump, ...	188,000 lbs.
11.	Weight of water supplied from jackets,	9,000 "
12.	Temperature of main feed-water, .	100 deg.
13.	" jacket-water, . . .	290 "
14.	Percentage of moisture in steam, . .	2.5

Additional Data based on the Above.

15.	Weight of one cubic foot of water at 60 deg. (w), ...	62.4 lbs.

16. Head corresponding to pressure in force-main $\frac{[80 \times 144]}{62.4} = H$, . .	184.6 ft.
17. Head corresponding to vacuum in suction-main $[4.8 \times 1.13] = h$, . . .	5.4 "
18. Total heat of 1 lb. of dry steam at 100 lbs. gauge-pressure, reckoned from 0 deg. Fahr., ...	1,216.5 th. un.
19. Total heat of 1 lb. of steam at 100 lbs. gauge-pressure containing 2.5 per cent. moisture, ...	1,194.6 "
20. Total heat of 1 lb. of steam at 100 lbs. gauge-pressure, containing 2½ per cent. of moisture, reckoned from temperature of main feed-water (100 deg. Fahr.), ...	1,094.6 "
21. Total heat of 1 lb. of steam at 100 lbs. gauge-pressure, containing 2½ per cent. of moisture, reckoned from temperature of jacket-water (290 deg. Fahr.), ...	902.2 "
22. Heat consumed by engine, $188000 \times 1094.6 + 9000 \times 902.2$ . . .	= 213,904,600 "

Applying these quantities in accordance with the duty formula we have:

Duty =

$$\frac{\left[(\overset{N}{140000} \times \overset{V}{12.5} \times \overset{w}{62.4}) - \overset{L}{546000}\right] \times \left(\overset{H}{184.6} + \overset{h}{5.4} + \overset{s}{10}\right) \times 1000000}{213904600}$$

$$= \frac{21730800000 \times 1000000}{213904600} = 100593268 \text{ foot-pounds.}$$

Second Case.—Jacketed compound direct-acting duplex engine. Jet condenser. Independent air-pump, which exhausts through a heater. Feed-water supplied by an independent

[illegible]-pump, which exhausts through the heater. Jacket-water returned to boiler without passing through heater.

1. Boiler-pressure by gauge . . . . 120 lbs.
2. Capacity of pump-displacement, one stroke (v) . . . . [illegible] cub. ft.
3. Number of strokes during trial (N) . . [illegible]
4. Pressure by gauge on force-main . . 100 lbs.
5. Vacuum by gauge on suction-main . 9.3 in.
6. Vertical distance between gauges (x) [illegible] ft.
7. Temperature of water in pump-well 80 deg.
8. Leakage of pump, determined by trial at rest (L) . . . . [illegible] lbs.
9. Form of pump-diagram . Rectangular (or F = unity)
10. Weight of water supplied by feed-pump . . . . [illegible] lbs.
11. Weight of water supplied by jackets [illegible] "
12. Temperature of main feed-water . . [illegible] deg.
13. " " jacket-water . . . [illegible] "
14. Percentage of moisture in steam . . 3 per cent.

Additional Data based on the Above.

15. Weight of 1 cubic foot of water at 80 deg. Fahr. . . . . 62.2 lbs.
16. Head corresponding to pressure in force-main, $\left(\frac{100 \times 144}{62.2} = H\right)$, . 231.5 ft.
17. Head corresponding to vacuum in suction-main, $(9.3 \times 1.13) = (h)$, . 10.5 "
18. Total heat of 1 lb. of dry steam at 120 lbs. gauge-pressure, reckoned from 0 deg. Fahr., ... 1,220.2 th. un.
19. Total heat of 1 lb. of steam at 120 lbs. gauge-pressure, containing 3 per cent. of moisture, ... 1,194.2 "
20. Total heat of 1 lb. of steam at 120 lbs. gauge-pressure, containing 3 per

cent. of moisture, reckoned from temperature of main feed-water (215 deg. Fahr.), ... 978.3 th. un.

21. Total heat of 1 lb. of steam at 120 lbs. gauge-pressure, containing 3 per cent. of moisture, reckoned from temperature jacket-water (280 deg. Fahr.), ... 912.2 "

22. Heat consumed by engine, $56000 \times 978.3 + 6400 \times 912.2$ . . . . = 60,622,880 "

Applying these quantities to the formula, we have:

Duty =

$$\frac{[(76000 \times 3.75 \times 62.2) - 354540] \times (231.5 + 8 + 10.5) \times 1000000}{60623880}$$

$$= \frac{4343115000 \times 1000000}{60622880} = 71641374 \text{ foot-pounds.}$$

Third Case.—Jacketed fly-wheel compound engine. Intermediate receiver fitted with reheater supplied with live steam. Main steam-pipe provided with separator. Water drained from jackets, reheater, and separator, into a closed tank under boiler-pressure, from which the water is supplied to the boiler by a small steam-pump. Jet condenser fitted with independent air-pump with exhausts through a heater to the atmosphere. Main supply of feed-water drawn from hot-well and fed to boiler by injector, discharging through the heater. The main feed-water and the auxiliary supply enter the same feed-pipe just before its connection to the boiler. The auxiliary pump exhausts through the heater. Number of plungers, two: diameter of each plunger, 19 in.; length of each stroke, 36 in.; diameter of each piston-rod, $3\frac{1}{2}$ in.

1. Boiler-pressure by gauge, ... 120 lbs.
2. Capacity of pump-displacement, one stroke,

$$\frac{\text{Each plunger} = \text{area } 19 - \tfrac{1}{2}\text{ area } 3\tfrac{1}{2}}{144} \times 3\,(V) = 5.806 \text{ cub. ft.}$$

3. Number of strokes of one plunger during trial (N), ...	280,000
4. Pressure by gauge on force-main, . .	80 lbs.
5. Vacuum by gauge on return-main, .	4.8 in.
6. Vertical distance between centres of gauges (s), ...	10 ft.
7. Temperature of water in pump-well, .	60 deg.
8. Leakage of plunger determined by trial at rest (L), ...	546,000 lbs.
9. Correction for admission of air into the pump (C), ...	.95
10. Weight of water supplied by main feed-pump, ...	188,000 lbs.
11. Weight of water supplied by auxiliary pump from the jacket, reheater, and separator-tank, ...	12,000 lbs.
12. Weight of water discharged from jackets, ...	7,000 "
13. Weight of water discharged from reheater, ...	2,000 "
14. Weight of water discharged from separator, ...	3,000 "
15. Temperature of water supplied from hot-well to injector, ...	110 deg.
16. Temperature of water discharged from injector and entering heater, . .	180 "
17. Temperature of feed-water leaving heater, ...	215 "
18. Temperature of jacket-water previous to entrance to tank, ...	290 "
19. Temperature of reheater-water previous to entrance to tank, ...	300 "
20. Temperature of separator-water previous to entrance to tank, ...	340 "
21. Percentage of moisture in the steam leaving separator, according to calorimeter-tests, ...	0.5 per cent.

22. Weight of 1 cubic foot of water at 60 deg. (w), ...	62.4 lbs.
23. Head corresponding to pressure in force-main, $\frac{80 \times 144}{62.4} = (H)$, . . .	164.6 ft.
24. Head corresponding to vacuum in suction-main, $4.8 \times 1.13 = (h)$, . .	81.1 "
25. Total heat of 1 lb. of dry steam at 120 lbs. gauge-pressure, reckoned from 0 deg. Fahr., ...	1,220.2 th. un.
26. Total heat of 1 lb. of steam at 100 lbs. gauge-pressure, containing 0.1 per cent. of moisture, = 1220.2 − [.005 × 867.5 (latent heat)], ...	1,215.9 "
27. Total heat of 1 lb. of steam at 120 lbs. gauge-pressure, containing 0.5 per cent. of moisture, reckoned from temperature of main feed-water corrected for heat derived from injector = 1215.9 − [110 + (215.9 − 180.6)] = 1215.9 − 145.3, ...	1,070.6 "
28. Total heat of 1 lb. of steam at 120 lbs. gauge-pressure, containing 0.5 per cent. of moisture, reckoned from temperature of jacket-water (290 deg.), = 1215.9 − 292, ...	923.9 "
29. Total heat of 1 lb. of steam at 120 lbs. gauge-pressure, containing 0.5 per cent. of moisture, reckoned from temperature of reheater-water (300 deg.), = 1215.9 − 302.1, ...	913.8 "
30. Heat lost by cooling of 1 lb. of the separator-water from its temperature in the boiler (349.9 deg.), to its temperature of discharge to tank (340 deg.), = 352.6 − 342.7, . .	9.9 "

31. Heat consumed by engine = (188000 × 1070.6) + (7000 × 923.9) + (2000 × 913.8) + (3000 × 9.9) . . . 209,597,400 th. un.

Applying these quantities in accordance with the duty formula, we have:

Duty =

$$\frac{\left[\left(\overset{C}{.95} \times \overset{V}{5.806} \times \overset{w}{62.4} \times \overset{N}{280000}\right) \times \left(\overset{H}{184.6} + \overset{h}{5.4} + \overset{s}{10}\right)\right] \times 100000000}{209597400}$$

$$= \frac{19274062080 \times 1000000}{209597400} = 91957544.$$

Where the lift is small, and especially if the quantity to be raised is large, the centrifugal pump, driven at high speed by fast-running engines, is generally employed. Such arrangements are also customary in steam-vessels. Centrifugal pumps are not economical under heavy lifts or where the cost of power is serious. Mr. Hansen gives the following, probably low, estimate for the power demanded by the best-known pumps of this type:

$$\text{I. H. P.} = \frac{10\,VH^{1.5}}{2 \times 33000} = \frac{VH^{1.5}}{6600};$$

where V is the number of gallons raised per minute through the height, H feet.*

The following table represents the results of trials of German naval ship-pumps of the centrifugal type†, as translated by Mr. Hansen:

---

* Engineering, February 22, 1889, p. 182.

† Busley: "Die Schiffsmaschine," 1883

WORK OF CENTRIFUGAL PUMPS.

Number of Experiment.	Name of Ship.	Duration of Trial. Minutes.	Seconds.	Steam-cylinder. Number.	Diameter.	Stroke.	Centrifugal Pumps. Number of pumps.	Diameter of disk.	Diameter of suction-pipe.	Diameter of delivery-pipe.	Lift. Suction.	Delivery.	Total.
1.	2.	3.	4.	5.	6.	7.	8.	9.	10	11	12.	13.	14.
					in.	in.		in.	in	in	ft.	ft.	ft.
1	Sachsen......	4	00.0	2	11.81	11.81	2	31.50	13.39	14.17	5.91	10.17	16.08
2	Würtemburg..	3	50.0	2	12.20	11.81	2	31.50	13.39	14.17	5.12	9.22	14.35
3	Leipzig......	2	57.5	2	19.57	12.36	2	43.70	12.36	12.36	11.65	2.62	14.27
4	Bayern.......	4	10.0	2	10.63	9.13	2	29.92	12.20	12.20	7.61	7.81	15.42
5	Bismarck.....	5	17.5	..	...	...	..	..	..	...	7.87	7.87	15.75
6	Blücher......	12	25.0	..	..	..	...	....	....	..	7.90	6.89	14.79
7	Gneisenau....	12	30.0	..	...	...	..	...	...	...	7.22	3.81	11.03
8	Moltke.......	6	45.0	..		....	.	.	...	..	8.53	5.91	14.44
9	Stosch.......	18	5.0	2	11.81	9.84	2	27.56	8.27	8.27	6.64	6.99	13.63
10	Stein........	5	41.5	2	10.63	9.37	2	27.56	12.40	12.40	8.27	3.28	11.55
11	Olga.........	2	8.0	2	9.06	9.45	2	27.56	11.81	11.81	9.35	1.48	10.83
12	Marie........	6	0.0	2	9.06	8.27	2	23.62	8.27	14.96	8.53	5.25	13.88
13	Blitz........	1	49.0	1	..	...	1	...	9.84	9.84	4.79	6.46	11.25
14	Möwe.........	8	0.0	2	6.30	5.51	2	12.60	4.72	4.72	4.59	4.59	9.19

Number of Experiment.	Quantity of Water Lifted. Total.	Per minute.	Steam Engine. Boiler-pressure.	Opening of Steam-valve	Revolutions per minute.	Mean Pressure on Piston.	Indicated Horse-power	Brake Horse-power.	Pump. Horse-power in water lifted	Useful effect.	Velocity of water in pipe.	Vacuum in suction-pipe.
1.	15.	16.	17	18	19.	20.	21	22.	23.	24.	25.	26.
	gals.	gals.	lb. per sq. in.			lb. per sq. in				per cent.	ft. per sec.	lb. per sq. in.
1	7,678	1919.5	28.45	1.00	260.0	13.07	45.08	...	9.37	20.8	5.25	3.56
2	6,754	1750.5	28.45	1.00	247.0	15.23	52.18	....	7.62	14.6	4.79 / 4.40	3.13
3	10,912	3692.6	22.47	1.00	211.[illegible]	12.05	95.29	....	16.00	16.8	3.94	3.41 / 3.84
4	7,920	1900.8	28.45	1.00	343.3	11.12	31.07	....	8.91	28.9	6.27	3.70 / 3.84
5	5,214	1004.0	28.45	1.00	350.0	....	.	14.10	4.73	33.7	3.31	
6	23,408	1885.8	28.45	1.00	400.0	..		16.37	8.45	51.6	6.20	
7	22,880	1830.4	28.45	0.22	337.0	....	...	12.13	6.00	50.0	6.04	
8	3,938	609.0	28.45	0.13	332.5	..		10.55	2.71	25.7	2.00	
9	20,282	1620.7	28.45	1.00	385.0	....	...	.	6.71	..	11.61	6.40
10	7,414	1310.5	27.02	1.00	324.5	..	..	...	4.59	..	4.17	4.12 / 7.11
11	5,434	2583.9	69.98	1.00	433.0	30.72	79.40	...	8.47	10.7	9.06	9.39
12	8,580	1430.0	51.20	1.00	370.0	12.38	84.46	..	6.03	24.6	10.26 / 3.15	3.84
13	1,738	957.0	42.67	0.25	325.0	12.37	10.75	...	3.26	30.3	4.82	1.72
14	1,276	159.5	99.56	1.00	420.0	..	....	...	0.27	...	3.51	

... Donkin System of trial has already ... the preceding chapter. The measure- ... charged from the condenser has been seen ... and most convenient and exact method ... thermodynamic study of the efficiency ... Professor Unwin, as just stated, has applied ... study of a pumping-engine of the Worth- ... the "equalizers" attached, forming what has ... "high-duty" type of that machine. The fol- ... reported results: *

... DIAGRAMS.

It was arranged that there should be an eight-hours' trial of the engines only, and a twenty-four-hours' trial of engines and boilers conjointly. The jacket drains were rearranged so that the jacket condensation could ... measured.

... Engines. The engines are compound, pumping a large volume of ... on a low lift. The high-pressure ... are ... in. in diameter, and the low-pressure pistons 54 in.

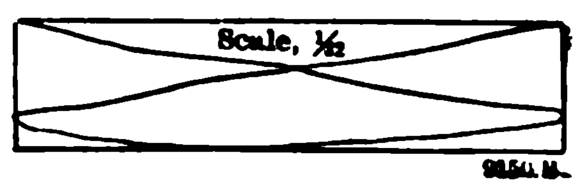
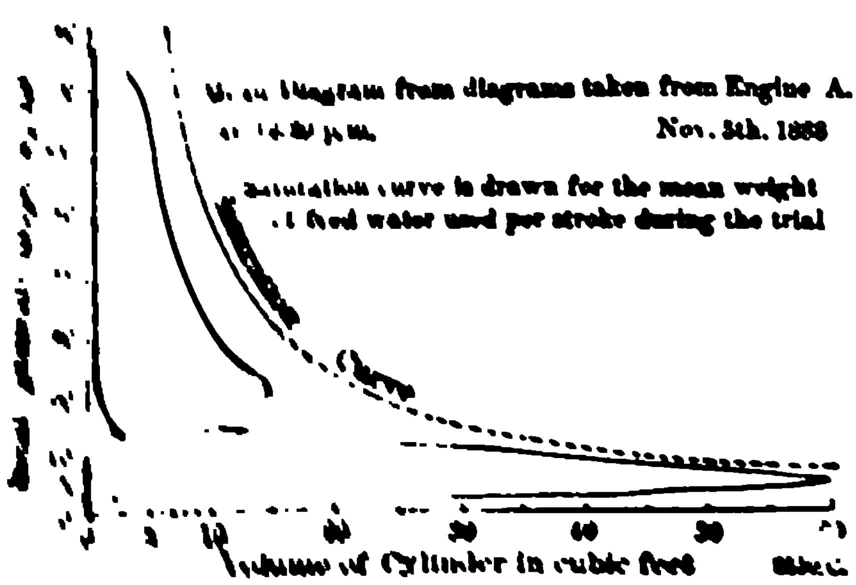


FIG. 127.—DIAGRAMS.

The stroke is variable, the maximum from cylinder-head to cylinder-head being 44 in. During the trials the stroke remained very constant and about 43 inches. The engines work rams 40 in. in diameter, and the same stroke as the steam-pistons.



F... ... MEAN DIAGRAMS.

Compensating cylinders absorb work during the first half of the stroke, and give it back during the second half. There are two to each engine, 11 in. in diameter, loaded by air-pressure to about 130 pounds per square inch. The pumps lift

* London Engineering, December 7, 1888.

water from a well communicating with the river and deliver it through two 3-ft. mains to the reservoirs, nine miles distant. The head during the trials measured by the difference of pressure in the suction and discharge pipes, was from 50 ft. to 65 ft. The head was measured by mercury columns fixed in the engine-house, communicating with the suction and delivery mains in accordance with the provision of the contract.

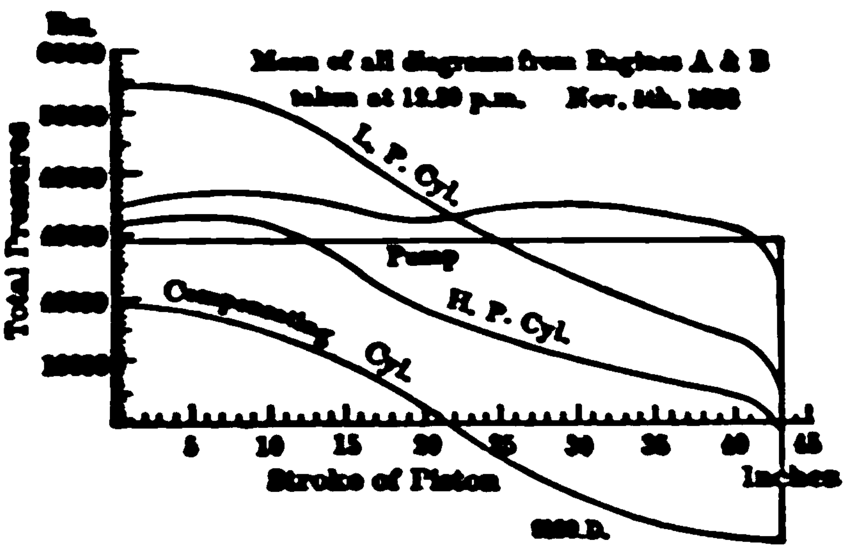


FIG. 129.—PRESSURE CURVES.

The suction-gauge com-

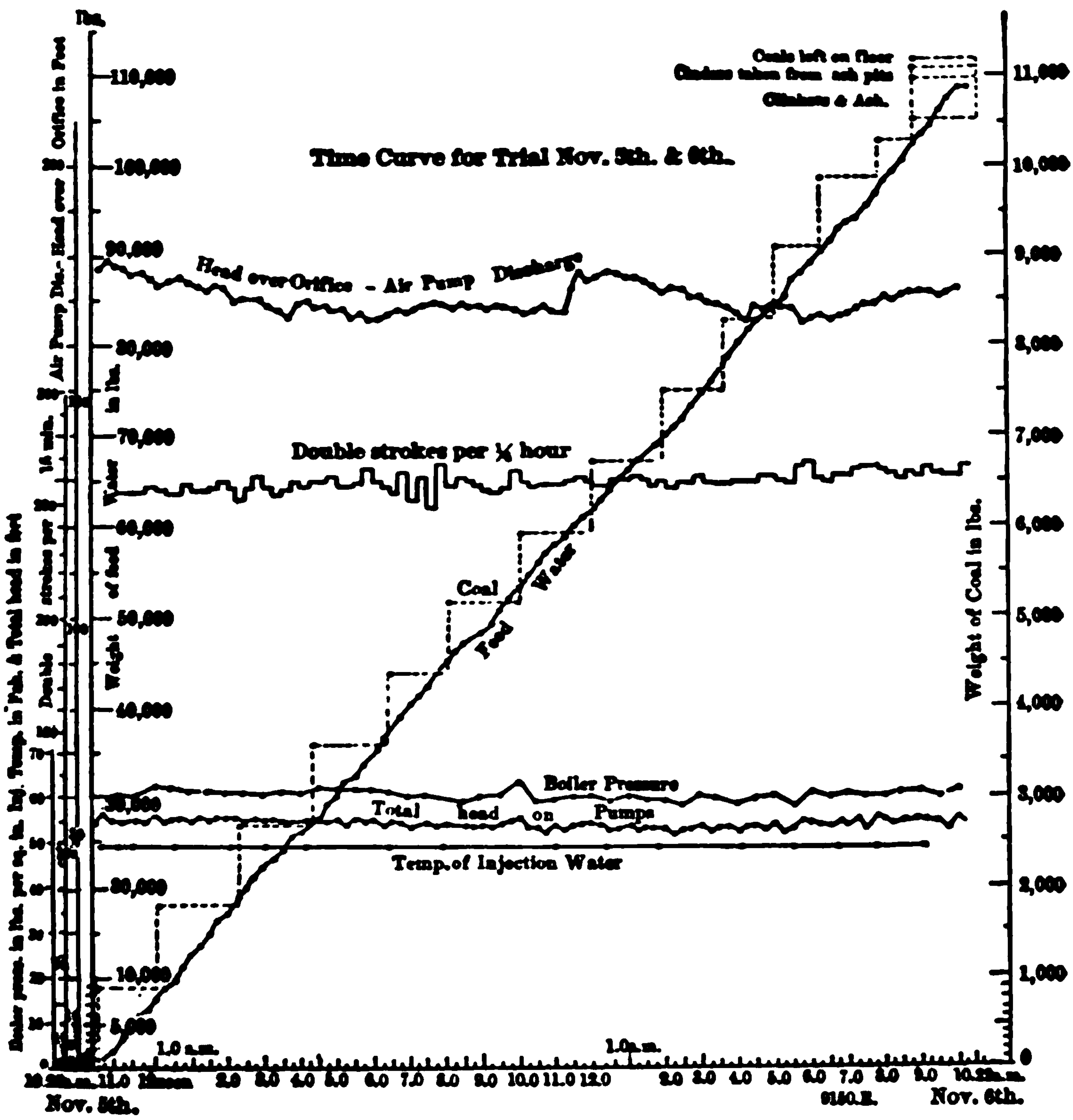


FIG 130.—CHART OF TRIAL.

municated with the suction-pipe just below the floor, and the pressure-gauge was set to give pressures reckoned from the floor-level. The sum of the mercury-gauge readings is taken as the effective lift. The pressure-gauge communicates with the delivery-main at a point beyond the stop-back valve. Consequently the resistance of that valve is reckoned as part of the engine friction, and is not credited to the useful work done by the pumps.

The engine-cylinders are jacketed, and the steam is also taken through a jacketed reservoir between the cylinders. The jacket-water was weighed. The condensers are injection condensers with horizontal air-pumps.

The pumps were very carefully measured, with the following results:

DIAMETER AND AREAS OF CYLINDERS AND PUMPS.

		Diameter at 60 deg F.	Diameter at 316 deg F.	Area of Piston	Area of Rod	Effective Area.	Means.
		in.	in.	sq. in.	sq. in.	sq. in.	
H. P. cylinder A........	Back	26.98	27.02	573.4	17.7	555 7	553.5
	Front	26 98	27 02	573.4	23 8	549.6	
H. P. cylinder B........	Back	27.02	27 06	575.1	17.7	557 4	
	Front	27 02	27 06	575.1	23.8	551.3	
L. P. cylinder A.......	Back	53 99	54 07	2296.2	7.0	2289.2	2285.1
	Front	53 99	54.07	2296.2	17 7	2278.5	
L. P. cylinder B........	Back	54.02	54 10	2298.7	7 0	2291 7	
	Front	54.02	54 10	2298.7	17 7	2281 0	
Pump plungers..........	Back	39 90	....	1250.0	16 8	1233.2	1241.6
	Front	39 90	...	1250.0	0	1250 0	

The Boilers.—The boilers were single-flued Cornish boilers. Three were used in the trial on October 29, and four in the trial on November 5 and 6. The boilers were 28 ft. in length and 6 ft. in diameter, with a single flue 3 ft. 6 in. in diameter for the greater part of the length.

During the trials of November 5 and 6 the length of the grate was 4 ft. 6 in. Hence the grate-area of the four boilers was 60 square feet. The feed-pipe was disconnected, and the safety-valves open on the idle boilers.

The coal was weighed under supervision on platform-scales, which had been tested, and the weights of coal brought into the house were from time to time again tested on a Denison balance.

Measurement of the Feed.—The feed was supplied from the delivery-main at a nearly constant temperature of 51 deg., the ordinary feed arrangements which supply the boilers with hot water from the jackets and hot-well being disconnected. The feed was delivered into a small gauge-tank with overflow-pipe provided with a float and counter. The capacity of this gauge-tank was determined three times by weighing the water; and the closely accordant measurements gave a mean value of 394 pounds for the capacity. No corrections are necessary for temperature, and no error is introduced by any possible difference of level or condition.

The feed-tank delivered by a stop-valve into another tank, from which a small Worthington feed-pump delivered the water into boilers.

The Worthington pump took its steam from the boilers in use, and exhausted into the tank, from which it pumped. The whole of the steam used was therefore recondensed and returned to the boilers.

Of the heat supplied by the boilers to work the feed-pump, nearly all was returned to the boilers. A small portion, viz., that due to the useful work of pumping and that lost by radiation from the tank, was no doubt lost. So far a small error telling against the main engines is introduced.

The water-level at the commencement of each trial in the boiler gauge-glasses was carefully observed, and the water-level was brought to exactly the same marks at the end of the trials. The time at which each tankful was supplied to the boilers was noted, and also the feed-water temperature. Pyrometer observations were made in the flues. Anemometer observations of the air supplied to each boiler were taken every half-hour during the twenty-four hours' trial, the anemometer having been previously tested.

Measurement of the Air-pump Discharge.—The air-pump

discharge was led into a wooden tank with stilling-screens. From this it was discharged through a sharp-edged circular orifice freely into the air. The diameter of the orifice was carefully tested after the trials, and the coefficient of discharge from similar orifices is known to be 0.599. The temperature and head over the orifice was noted every 5 minutes in the first trial and every 7½ minutes in the second. The temperatures relied on in this report were taken by a fixed zero thermometer, with open scales, recently verified at Kew.

Measurement of Length of Stroke.—As the stroke is variable, an arrangement of indicating-fingers was attached to each engine, and the length of stroke on each engine was noted every quarter of an hour.

Indicated Power.—The indicated power was taken by four Richards indicators, chosen because they give fairly large diagrams. These indicators were sent to Kensington after the trials and tested under steam, against a steel-tube pressure-gauge recently made, and specially tested by Messrs. Schaffer and Budenburg. No important error was found at any part of the scale with any of the springs. But with the light springs of the low-pressure-cylinder indicators there was a little frictional sticking or else a little slackness of the parallel-motion joints, which under a steady pressure introduced a small uncertainty of indication at one or two points in the range. Probably this would be less still when the indicator-piston was in motion, as when drawing a diagram. The indicator-pipes were large and were clothed. Diagrams were taken every half-hour from all the cylinders.

Trial of Engines.

This was a twenty-four-hours' trial, the coal-consumption being measured, as well as the efficiency of the engine. The engines had been started in the morning, but, before beginning, the fires were cleaned and all ashes removed; also all coal was swept from the boiler-house floor. Four boilers were used, and the fires were not drawn; but the condition of the fires was nearly identical at the beginning and end of the experiment.

The trial commenced at 10.22 A.M. on the 5th and ended exactly at 10.22 A.M. on the 6th.

The barometer varied a little during the twenty-four hours, the mean being 29.78 in. (corrected), corresponding to 14.627 lbs. per square inch. The temperature of the injection varied from 48.6 deg. Fahr. to 49.5 deg. Fahr., the mean being 49.2 deg. Fahr. The mean boiler-pressure was 60.29 lbs. per square inch (74.92 lbs. per square inch absolute). The mean vacuum shown by the mercury-gauge on the engine was 27.76 in., or 13.63 lbs. per square inch. The total head of water on the pumps was about 55 ft. at starting and 53.5 ft. at the end of the trial; the mean head was 53.68 ft. The air-pressure in the compensating air-vessel varied from 118 lbs. to 122 lbs. per square inch (above atmosphere).

Speed and Length of Stroke.—The speed was remarkably constant, the mean speed being 17.282 double strokes per minute. The engines made 24,886 double strokes in the twenty-four hours. The length of stroke was even more constant than in the previous trials. The mean length of stroke was 43.06 in. for engine A and 43.05 for engine B.

Indicated Horse-power.—The reduction of diagrams taken every half-hour during the first eight hours, and every hour afterwards, gave the following results. The variation of the diagrams was very small:

		Indicated Horse-power.		
Engine A.—	H. P. back	31.662		
"	L. P. "	31.145	62.807	
"	H. P. front	34.176		128.668
"	L. P. "	31.685	65.861	
Engine B.—	H. P. back	35.856		
"	L. P. "	28.073	63.929	
"	H. P. front	35.236		126.849
"	L. P. "	27.684	62.920	
Total indicated horse-power of both engines, ...				255.517

The Pump.—The mean lift was 53.68 ft.; mean length of stroke, 3.5879 ft.; number of strokes per minute, 17.282. Hence the pumps lifted 13,407 gallons per minute, or 804,396 gallons per hour, or 19,305,504 gallons in the twenty-four hours. The pump horse-power is 217.06. Consequently the mechanical efficiency of the engines and pumps is 0.8495, slightly greater than in previous trials.

The Feed and Jacket Water.—The feed-water had a mean temperature of 51.07 deg. The total feed-water used was 108,537.4 lbs., or 4522.39 lbs. per hour. The jacket-water was measured for six hours on the 5th and for one hour on the morning of the 6th. The rate of discharge appeared to be the same. The amount of drainage from the jackets was 706 lbs. per hour. Consequently, reckoned per indicated horse-power per hour, the quantities were:

Total feed (at 51.07 deg.) per indicated horse-power per hour,	17.700
Jacket condensation,	2.763
Used in cylinders,	14.937

Air-pump Discharge.—The mean head over the orifice was 1.7033 ft., and the mean temperature 74.965 deg.* The total air-pump discharge was 2586 lbs. per minute, or 2522.4 lbs. of injection-water and 63.6 lbs. of condensed steam.

Heat Rejected by the Engine per Indicated Horse-power per Minute.—The heat required to raise the whole air-pump discharge from 49.2 deg. to 74.965 deg. We get for the heat rejected 260.7 thermal units per indicated horse-power per minute. This is Donkin's coefficient. The more accurate estimate of the heat rejected is as follows:

	Thermal Units.
Heat due to 2522.4 lbs. of injection-water per minute raised from 49.2 deg. Fahr. to 74.965 deg. Fahr.,	64,990

* Another set of readings, with another thermometer, gave this temperature 75.13 deg. Fahr.

Heat due to 63.6 lbs. of feed-water raised from 51.07 deg. Fahr. to 74.965 deg. Fahr., ...	1,519
Heat due to 11.78 lbs. of jacket-water raised 256.3 deg. Fahr., ...	3,020
	69,529
Heat rejected per indicated horse-power per minute, .	272.1
Add converted into work, ...	42.7
	314.8

Which neglects the loss by radiation.

Heat used, reckoned from the Boiler-pressure.—The total heat of the steam, considered dry, reckoned from the feed-temperature at the mean boiler-pressure is 1156.5 thermal units per pound. Consequently the heat delivered from the boiler to the engine was 341.1 thermal units per indicated horse power per minute. The difference between this and the previous estimate, of 314.8, represents loss by radiation, error due to the presence of priming-water in the steam, and errors of observation.

If we suppose the jacket-water pumped into the boiler at the temperature of the steam (as it returns to boilers in a closed circuit), and the rest of the feed taken from the hot-well, thus removing the abnormal conditions which were present in the trial, 17.8 thermal units, or 5.2 per cent. of the heat used per indicated horse-power per minute, would be saved. Then the heat required by the engine would be, in normal conditions of working, 323.3 thermal units per indicated horse-power per minute.

The following table gives the results:

Double strokes per minute, ...	17.282
Boiler-pressure, ...	60.29 lbs. per sq. in.
Feed-water per minute, ...	75.37 lbs.
Jacket-drains per minute, ...	11.77 "
Temperature of steam, ...	307.36 deg. F.
Pressure on pump, ...	23.26 lbs. per sq. in.
" in compensators, ...	120 " "

Mean pressure in high-pressure cylinders,	32.92	lbs. per sq. in.
Mean pressure in low-pressure cylinders,	6.905	" "
Temperature of injection, ...	49.2	deg.
" " air-pump discharge, . .	74.965	"
Head over orifice, ...	1.7033	ft.
Air-pump discharge per minute, . .	2586	lbs.
Injection-water per minute, ...	2522.4	"

Heat passing through engine per indicated horse-power per minute:

Thermal units from boiler in saturated steam through cylinders from feed-temperature, .	287.8
Latent heat of jacket steam, ...	41.45
	329.25
Heat rejected in air-pump discharge, .	260.24
Converted into work, ...	42.75
Radiation and error, ...	36.26
	329.25

Indicated horse-power, ...	255.517	
Pump, "	217.06	
Mechanical efficiency, ...	.8495	
Feed per indicated horse-power per hour through cylinders, ...	14.937	lbs.
Feed per indicated horse-power through jackets, ...	2.763	
Piston speed per minute, ...	124	ft.

Measurement of Coal used.—The floor having been swept clean, the coal was brought in in quantities of about 8 cwt., and the time of finishing each lot was noted. The ash-pits were cleaned before the trial, and afterwards nothing was removed till the end. The fires were cleaned before the trial began, and again at 4 A.M. on Tuesday morning. The fires were not touched at the end of the trial, but the ash-pits were immediately cleaned, and the whole of the ashes were treated thus:

First the clinkers were separated and weighed. The rest of the ashes were sifted through a sieve with ½ in. mesh. All that

passed through the sieve is treated as incombustible ash, although probably one-third of it is unburned carbon. What did not pass through the sieve is treated as unburned fuel. Analysis in similar cases has shown that the cinders retained by the sieve are almost entirely carbon.

The coal account, then, stands thus:

	Pounds.	Pounds.
Gross weight of coal brought into boiler-house,		11,180
Left on floor at end of trial, ...	99	
Cinders sifted out of ashes, ...	132	231
Total coal used, ...		10,949
		= 456.2 lbs. per hour.

	Pounds.
The residue consisted of clinkers, ...	66
Incombustible ashes, ...	366
	432

The clinkers and ashes amount to 3.9 per cent. of the coal used.

The rate of combustion was 7.24 lbs. of coal per square foot of grate, or 0.19 lbs. per square foot of heating-surface, per hour. The coal used per indicated horse-power per hour was 1.785 lbs., a very good result, as the feed was supplied at 51 deg. Fahr., and the rejected heat from the jacket-drains wasted. The evaporation was 9.914 lbs. of water from 51.07 deg. at 307.36 deg. per pound of coal including clinkers and ashes. This corresponds to an evaporation of 11.867 lbs. per pound of coal from and at 212 deg.

Calorimetric Value of the Coal.—The heating power of the coal has not been directly determined, but good Welsh coal is known to contain about 89 per cent. of carbon and 4 per cent. of hydrogen, the rest being oxygen, nitrogen, and ash. The calorimetric value of such a fuel is

$$14500\{0.89 + 4.28 \times .04\}$$
$$= 15387 \text{ thermal units per pound.}$$

But this is reckoned for a dried sample of coal, and makes no allowance for the latent heat of the steam produced in combustion. There would be produced by combustion 0.36 lbs. of water per pound of coal; and the latent heat of this would be 348 thermal units: so that the available heat of a pound of dry coal would be 15,039 thermal units. The coal as taken from the yard would contain at least 1 per cent. of moisture, so that the available heat of 1 lb. of the coal as weighed and used would be:

	Thermal Units.
Heat due to 0.99 lb. of coal, ...	14,888
Less latent heat of 0.01 lb. of water, ...	10
	14,878

Available heat, 14,878 thermal units per pound of coal as weighed and used. Taking this value, the total heat due to the combustion of the coal is 26,557 thermal units per indicated horse-power per hour, or 442.6 thermal units per minute per indicated horse-power. Of this 341.1 has been shown to be delivered to the steam. There remain 101.5 thermal units per indicated horse-power per minute to account for as losses in the boilers. The efficiency of the boilers is 0.77. The coal gave to the steam 11,466 thermal units per pound of coal used.

Anemometer Observations. — Observations at each boiler every half-hour gave the following volumes of air entering per minute in cubic feet at the temperature 79.5 deg. of the boiler-house.

Boiler, ...	J	K	L	M
Quantity of air in cubic feet per minute, .	420	438	486	360

Hence the total quantity of air used was 1704 cubic feet per minute, or 225 cubic feet per pound of coal. The weight of the air used was 7489 lbs. per hour, or 16.42 lbs. per pound of coal. As the coal requires nearly 12 lbs. per pound for perfect combustion, the quantity of air used was moderate.

The mean temperature of the flue from the pyrometer observations was 422 deg.

Tabulating the results stated, we get:

	Per Hour.	Per Indicated Horse-power per Hour.
	lbs.	lbs.
Coal used, ...	456.2	1.785
Air used, ...	7,489.0	29.310
	7,945.2	
Less ashes and clinkers, ...	18.0	
Total weight of furnace gases, . .	7,927.2	31.03

Heat Used and Lost in Boilers.—The thermal units of heat developed in the furnaces were applied thus:

	Thermal Units per Indicated Horse-power per Hour.	Per Cent.
Total heat due to coal used, ...	26,557	100
Given to steam, ...	20,466	77.1
Carried off in furnace gases, ...	2,657	10.0
Probable loss due to opening firedoors to stoke,	265	1.0
Due to carbon in ashes, ...	284	1.1
Radiation and unaccounted for, ...	2,885	10.8

If there was any priming-water, the heat given to steam would be less. On the other hand, probably, the losses due to moisture in the coal and to air entering the furnaces during stoking are underestimated.

Duty of the Engines.—The work done by the engines during the twenty-four hours' trial was 106010000 foot-pounds per 112 lbs. of coal.

It has been stated that for the purposes of the trial the ordinary conditions of the engines were altered, and heat rejected which is ordinarily used. Correcting for this, the duty of the engines in normal conditions of work must be 111.5 millions according to the results of the trial.

To accompany this report, drawings are sent as follows:

Drawing 1.—A mean diagram (see Fig. 128) drawn from

the diagrams taken on engine A at 12.30 P.M. On this has been plotted a saturation-curve for the mean speed per stroke during the trial. Since the indicated power varied so little, this saturation-curve must be very approximately the true curve for the actual diagrams. The re-evaporation during the stroke is very marked, as was to be expected from the large jacket condensation.

Drawing 2.—Mean diagrams (see Fig. 129) from all the diagrams of both engines taken at 12.30 P.M. are plotted so as to show the effective thrust of the engines at each point of the stroke. A curve of cosines is drawn giving the ± thrust of the compensators. Combining this with the engine-diagram, the resultant thrust is obtained. The effect of the inertia, however, is neglected. It will be seen that the resultant thrust is remarkably uniform, and probably the effect of the inertia of the moving pistons and plungers is to increase the uniformity of this thrust.

Drawing 3.—The principal observations taken during the trial have been plotted in this diagram (see Fig. 130) with time abscissæ. The diagram shows the general regularity of the working of the engines during the trial.

104. Gas-engine Trials, and Reports on their results, are substantially similar to those for steam-engines: but here engine and boiler are one, and all heat-energy is generated by the combustion of the fuel within the working cylinder; this compelling special methods of measuring it. The most complete and fruitful example of such a trial is described in the report an abstract of which follows.* Some of its deductions are given in another chapter (§ 84).

The report on these trials, by Messrs. Brooks and Steward, embodies a very careful examination, both theoretically and experimentally, of the performance of that form of motor, made with a view to determine not only the actual efficiency and economy of the machine, but also the extent and the proportions of the losses met with in its ordinary operation. A series of results had been obtained during investigations made

* Van Nostrand's Magazine, 1883.

under the direction of the Author, and which included trials of various forms of gas engine, in which the distribution of heat in useful and lost forms of energy was determined with care.

In these cases, the consumption of water-gas varied from 21.2 to 23.4 cubic feet per hour and per horse-power in engines of 6 or 7 indicated horse-power, up to 23.5 to 24.5 with engines of two horse-power or less. The friction of mechanism ranged from 4 to 5 per cent. of the total energy of combustion, and from 40 per cent. power in the smaller to 20 per cent. in the larger engines; the waste at the exhaust was from 12 per cent. of the total heat of combustion in the small to 24 per cent. in the large engines, and from 100 to 200 per cent. of the quantity transformed into useful work.

The water-jacket carried off from 45 to 55 per cent. of all the heat supplied by the combustion of the gas. For example: the distribution of the heat of combustion, in one case worked for the Author by his senior assistant, Mr. Cartwright, was as below:

Useful dynamometric work, ...	14.27
Work of the pump, ...	0.42
Friction of mechanism, ...	4.10
Lost in exhaust, ...	23.55
Ditto in water-jacket, ...	46.90
Radiation, etc., ...	10.76
Total heat supplied, ...	100.00

This engine developed seven horse-power at the brake, and indicated 8.9 H. P., consuming 21.2 or 27.6 cubic feet of gas, accordingly as the indicated or the dynamometric horse-power was made the basis of the calculation.

It seemed to the Author desirable that this method of investigation should be further developed, and that a comparison of the actual with the thermodynamic performance of the gas-engine should be made, systematically determining all the data needed to make such a comparison complete, and as nearly exact as possible. This work was undertaken by Messrs.

Brooks and Steward. Messrs. Schleicher, Schumm & Co., the builders of the engine, prepared one of their 10 horse-power Otto engines for trial, and forwarded it from Philadelphia. This engine was set up, as described in the report, and connected as shown in the plan, Fig. 131. The American Meter Company supplied meters, including one of unusual size, for the purpose of measuring the air, as well as the gas supplied, a measurement never before undertaken, so far as the Author is informed. The Meter Company also afforded all needed facilities for testing the meters, before and after the trials.

The results of the investigation are given with all necessary detail in the body of the report. It will be seen, as one result of the precaution taken to measure the supply of air, that the relative volumes of air and gas are found not to be precisely determinable from figures obtained without the use of an air-meter. This precaution was also found to have value as permitting a correct determination of the effect of varying the supply of air and of gas independently, either with or without change of proportions (Section 5).

The determination of the proper proportion of air to gas is important and interesting (section 9); and the comparison of the lines of the indicator with theoretic curves is still more interesting and novel, as well as very instructive (sections 12 and 13). The fact that combustion is progressive, even into the expansion period, is probably here, for the first time, exhibited by direct investigation (section 15). The analysis of the efficiency of the engine affords a means of making a comparison of the thermodynamic with the actual efficiency of this class of heat-engine. It is seen that the total heat accounted for thermodynamically, consisting of that transformed into work and that expelled with the exhaust, amounts to 34 per cent. of the total heat actually supplied, and that the other wastes, by way of the water-jacket and otherwise, amount to 66 per cent. The thermodynamic efficiency is therefore about $\frac{17}{33}$, and the actual efficiency $\frac{17}{100}$, or 52 and 17 per cent., respectively.

(1) *The Engine and Accessories.*—Below are the dimensions

of the principal parts of the engine, and of such accessories as are involved in the calculations:*

Stroke	14	in.	(356 mm.)
Diameter of piston	8.5	"	(216 ")
" " piston-rod	1.75	"	(44 ")
" " connecting-rod (crank end)	2.5	"	(63 ")
" " connecting-rod (piston end)	2.25	"	(57 ")
" " crank-shaft	4	"	(102 ")
" " fly-wheels	66	"	(1680 ")
" " brake-pulley	30	"	(762 ")
Length of brake-arm	16.5	"	(420 ")
Weight of both fly-wheels	1650	lbs.	(750 kilos)

Clearance (compression-chamber) 38 per cent. total cylinder volume.

The ground-plan (Fig. 131) shows the general arrangement of the apparatus employed, A meter was used to measure the gas used by the engine, inclusive of the igniting flames. As the engine takes gas suddenly and at intervals, it is necessary to insert a flexible rubber bag in the gas-supply-pipe between the meter and engine, to act as a gas reservoir, and so relieve the meter of all strain.

The air was also measured by a three-hundred-light meter, and a pair of large rubber bags was inserted in the air-pipe. A small fan-blower kept these bags constantly filled, the pressure being controlled by a check-valve in the pipe near the blower. A three-way cock in the air-pipe under the engine allowed the air to be taken either through the large meter or directly from the room.

The water required for the jacket was measured by a meter placed near the gas-meter, and its temperature, both before entering and after leaving the water-jacket, was measured by a standard thermometer.

The three meters were tested before they were put in place.

A pyrometer was placed in the exhaust-pipe as near as pos-

* For complete description see U. S. Letters Patent, No. 196,473, dated October, 1887.

sible to the engine, giving the mean temperature of the discharged gases.

For the purpose of measuring the work of the engine a Prony brake was employed.

The indicator was placed directly upon the cover of the exhaust passage.

A speed-counter was attached to the link moving the slide-valve, to record the number of double revolutions.

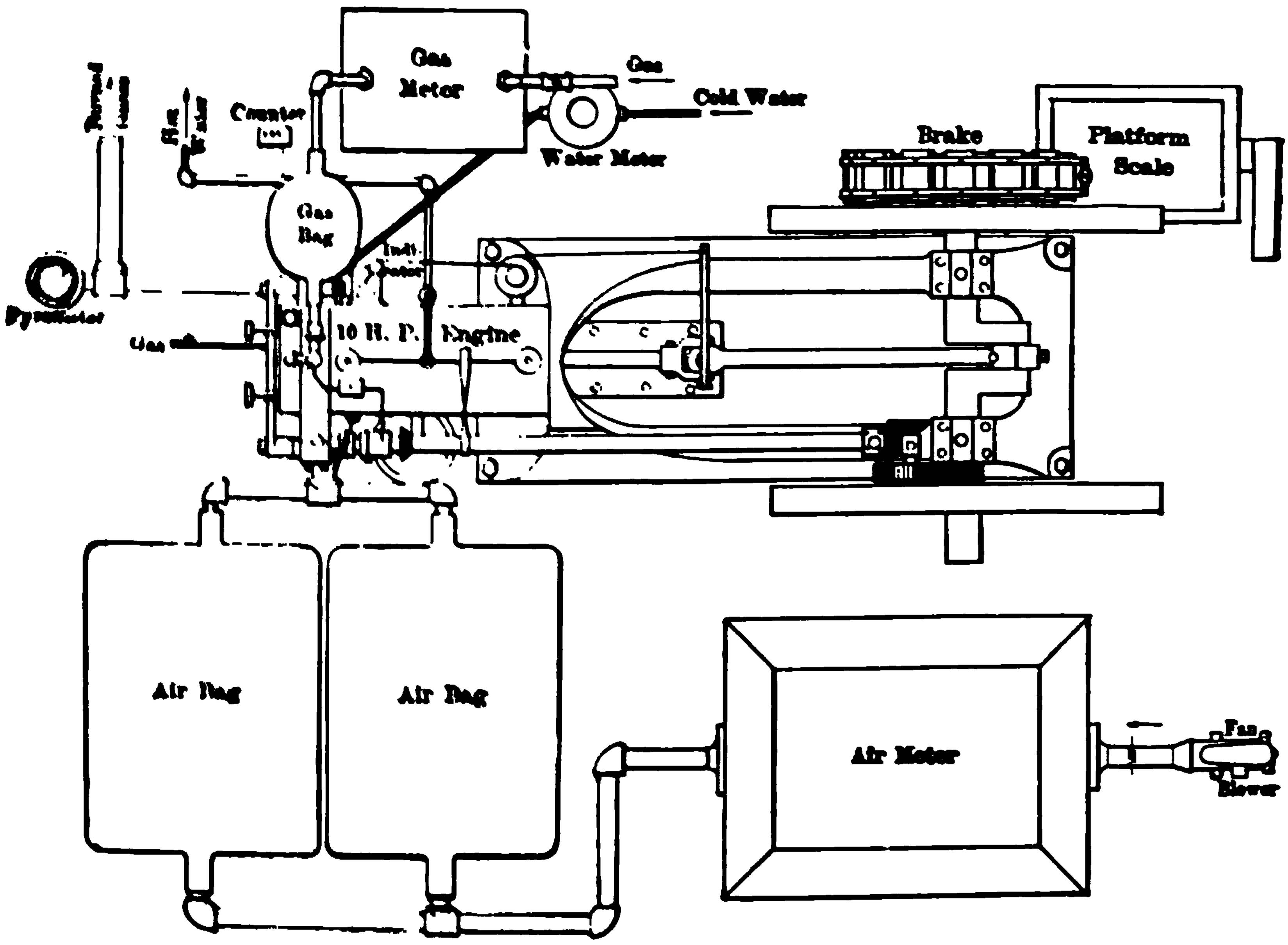


FIG. [illegible]. PLAN OF ARRANGEMENT FOR GAS-ENGINE TRIAL.

(2) *Summary of Tests.* The observations were usually made at intervals of five minutes during the tests, oftener when any marked variation was noticed. The gas-pressure was about 30 millimeters (1¼ inches) water column, and the air-pressure, when the large meter and blower were used, averaged 50 millimeters (2 inches).

The "horse-power in gas burned" was calculated from the analysis of gas, and is the dynamic equivalent of the heat capacity of the gas. The "horse-power lost by exhaust" and

GAS-ENGINE TRIAL.

	At Varying Powers.									At Full Power.						
Number of Test.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Date	5/29	5/28	5/30	5/30	5/30	6/4	6/4	6/4	5/29	5/28	6/4	6/4	6/4	6/4	6/4	6/29
Duration in minutes	15	15	30	14.5	30	10	15	10	60	30	30	30	30	30	30	30
Gas { cubic feet	12.5	17.5	63	41.5	99	28.5	46.5	34.7	249	109	115	122	105	114	115	11705
Gas { liters	355	495	1785	1175	2805	805	1310	980	7050	3090	3255	3460	2975	3227	3255	3320
Air { cubic feet	385	—	—	—	—	277	416	283	1538	732	806	765	796	831	816	806
Air { liters	10900	—	—	—	—	7850	11800	8000	43550	20740	22850	21670	22540	23510	23040	22850
Ratio of air to gas	—	—	—	—	—	—	—	—	—	6.6	6.9	6.2	7.4	7.1	7.0	6.86
Amount of water through jacket { cu. ft.	.94	4 6	4.3	4	10.8	2.5	3.75	2.45	24.3	7.8	11.5	6.0	6.65	4.45	8.75	11.8
Amount of water through jacket { liters	26.5	131	122	113	306	71	107	69.5	690	221	325	170	188	127	248	334
Temperature of water in °C. { entering	22	22	21	21	21	22	22	22	22	22	22	22	22	22	22	23
Temperature of water in °C. { leaving	49	29	51.7	44.5	42	49.5	48	51	45.7	59.7	48	80	76	80	57	49
Number of double revol.	1252	1247	2445	1184	2426	809	1212	801	4634	2100	2325	2299	2291	2461	2404	2371
Number of rev. per minute	167	166	163	163	162	162	162	160	154.5	140	155	153	153	164	160	158
Number of explosions	235	352	1224	788	1820	607	970	716	4620	Same as the number of double revolutions.						
Ratio of expl. to double revs.	.19	.3	.5	.66	.75	.75	.8	.89	.997	1	1	1	1	1	1	1
Average weight on brake-arm { lbs.	0	0	64	106	125	107	125	150	183	197	188	191	178	178	176	195
Average weight on brake-arm { kilos.	0	0	29	48	57	49	57	68	83	89	85	87	81	81	80	88
Mean effec. press. (atmos.)	4.2	3.3	4.4	4.3	4.1	4	3.9	4	4.2	4.2	4.1	4.1	3.9	3.9	3.8	4.04
Temp. of exh'st gases (°C.)	163	121	299	349	371	371	366	402	416	399	429	432	427	427	432	430
Indicated horse-power	2	2.55	5 4	7.06	7.6	7.3	7.6	8.6	9.4	8.4	9.6	9.5	8.9	9.7	9.6	9.6
Effective (brake) H. P.	0	0	2.73	4.5	5.3	4.5	5.4	6.3	7.6	7.2	7.8	7.8	7.1	7.7	7.5	8.1
H. P. lost by exhaust gases	2.3	—	—	—	—	6.7	6.8	7.6	7.6	6.9	7.9	7.8	7.7	8.1	8	8
H. P. lost by water-jacket.	6.6	8.1	11.5	16.8	19.8	18	17	18.7	25.2	25.7	26.2	30.5	31.2	22.4	26.8	26.9
Horse-power in gas burned	12.2	17.1	30.9	42.1	48.5	41.9	45.6	51	61	53.4	56.4	59.8	51.5	55.9	56.4	57.6
Gas per indicated H. P. per hour { cu. ft.	25	27	23.3	24.6	26	23.4	24.5	24.2	26.3	26	24	25.6	23.6	23.5	24 2	24.5
Gas per indicated H. P. per hour { liters	708	765	660	697	736	663	694	685	745	735	680	725	668	665	685	694
Gas per effective H. P. per hour { cu. ft.	—	—	46.1	38.4	37.4	38	34.4	33	32.7	30	29.5	31.1	29.5	29.4	30	29.1
Gas per effective H. P. per hour { liters	—	—	1310	1090	1060	1080	970	940	930	850	835	892	835	832	850	825

"by water-jacket" were calculated from the specific heats of the discharged gases and of water. The indicated horse-power was computed in the ordinary way from the number of explosions and the mean effective pressure. An allowance was made for the area between the exhaust and admission lines, representing work done in expelling the burned gases, and in drawing in the fresh charge, equivalent to a little more than one tenth of an atmosphere mean effective pressure.

The figures given for gas consumed do not include the amount burned by the igniting flames. An allowance of 7 cubic feet (200 liters) per hour will cover this.

(3) *Friction of Engine.*—The difference between the indicated and the actual work gives the friction of engine. The average of all the tests at full power is 18.6 per cent. friction, a remarkably good result.

(4) *Gas Consumption.*—The consumption of gas per horse-power per hour is large, because of the poor quality of the gas. The calculations show that only 5495 calories can be obtained from the combustion of 1 cubic meter. For average quality illuminating-gas 6000 calories is usually taken as a fair figure.

The following table shows the effect of various allowances:

GAS CONSUMPTION (HOURLY).

Engine Exerting Maximum Power.

		Average.	This amount reduced to 0° C.	Equivalent quantity of ordinary gas.	This amount reduced to 0° C.
Gas per indicated H. P.	cubic ft.	24.5	22 5	22.4	20 6
	liters.	694	637	634	583
Gas per effective H. P.	cubic ft.	30.1	27.7	27.6	25.4
	liters.	853	784	781	719

The table gives the results of tests made when there was not sufficient resistance to make the engine take gas every time. This is the ordinary condition of running, for it is well to have a reserve of power for the governor to call upon. The gas consumption with varying power is seen to be nearly constant per indicated horse-power.

(5) *Ratio of Air to Gas.*—The ratio of air to gas was found to be about seven to one, when the engine was working most economically. The ratio is commonly obtained from a measurement of the gas consumption alone, the air being reckoned as the volume of the piston-displacement, less the measured amount of gas. This is not an accurate method.

When the proportion of air is increased by partly closing the gas-valve the explosion line is much more inclined, the mean effective pressure is less, as is also the indicated horse-power. The gas consumption per effective horse-power becomes considerably greater.

(6) *Temperature of Water jacket.*—The amount of heat carried off by the water-jacket is about half of the total heat of combustion. The cylinder is kept cool by a plentiful supply of water. The quantity of heat carried away is greater than when less water is used.

But a comparison of the results fails to show any marked difference in either the indicated or the actual work caused by varying the temperature of the water-jacket.

(7) *Disposition of the Heat.*—Heat is disposed of in a gas-engine thus:

(1) As indicated work, including useful work and friction.
(2) In the hot expelled gases.
(3) In the water-jacket.
(4) In radiation, etc.

Taking the figures directly from test 19, but making allowance for probable error in the figure for water-jacket, it is found that

(1) = 1.33	calories or	17	per cent.;
(2) = 1.18	"	$15\frac{1}{2}$	"
(3) = 4.00	"	52	"
(4) = 1.18	"	$15\frac{1}{2}$	"

the sum, 7.69 calories, being the total heat of combustion of the gas.

The method of calculating the specific heat and thermal contents of the gaseous products of combustion has been already described.

Professor Unwin, making a trial of an Atkinson gas-engine, a form in which a variable stroke of piston permits extending the expansion of the exploded gases beyond their original volume at atmospheric pressure, obtained the results tabulated on p. 445.

For the efficiency of the mechanism we get

	Efficiency.
Trial V. Maximum load, ...	0.093
" I. Normal full power, ...	0.879
" II. Two-thirds full power, ...	0.800

They show that there is no exceptional friction in the peculiar arrangements of link work adopted.

The consumption of gas in the trials was as follows:

	Brake Horse-power.	Total Gas used in cubic feet per hour.	Gas used in cubic feet per hour per Brake Horse-power.
Trial V.	5.255	116.20	22.11
" I.	4.889	110.04	22.51
" II.	3.320	90.58	27.24
" III.	1.642	60.82	37.04

Also, without load, the engine used 37.42 cubic feet per hour.

The number of ignitions per minute was as follows:

Full power	....	147 (every revolution).
$\frac{2}{3}$ "	....	119
$\frac{1}{3}$ "	....	75
No load	....	54 (every third revolution).

The governor has therefore ample control, and the speed is regular, even with no load.

Studying, first, Trial I, which was the longest trial, and the one in which normal conditions of working were most nearly

GAS-ENGINE TRIAL.

Duration of Trial.	Load on Brake in lbs.	Revolutions of Engine per minute.	Gas used by meter in cu. ft. per hour.	Gas used per hour corrected for probable error of meter.	Jacket-water.		Pressure in Cylinder lbs. per sq. in. above Atmosphere.			Indicated Horse-power.	Brake Horse-power.	Gas used per hour in cu. ft.	
					Quantity in lbs per minute.	Rise of Temp. ° F.	Mean Initial.	Mean Terminal.	Mean Effective.			Per Indic. H. P.	Per Brake H. P.
H. M. H. M.													
I. 11 00 to 12 00	68.10	147.80	111.38	110.04	2.09	106.9	145.9	15.0	33.32	5.563	4.889	19.78	22.51
II. 12 15 to 1 00	45.86	149.31	91.68	90.58	2.08	99.3	97.5	15.4	30.95	*4.160	3.326	21.78	27.24
III. 2 30 to 3 05	22.54	149.9	61.56	60.82	2.04	65.2	....	...	...	....	1.642	....	37.04
IV. 3 20 to 3 55	0	150.1	37.42	36.97	2.01	52.05	....	....	....	....	....	....	....
V. 4 05 to 4 35	74.08	146.04	117.6	116.2	2.00	†88.6	132.1	15.5	35.22	5.811	5.255	20.00	22.11

* From a five-minute count, there were 119 ignitions per minute.

† The temperature of jacket-water was rising, and did not become constant.

present. In one minute 1834 cubic feet of gas were used, which at 628.7 thermal units per cubic foot would furnish altogether 890,000 foot-pounds of work, if it could all be rendered available.

For this there is obtained:

	Foot-pounds.
On brake 4.889 h.p. or	161,337
Engine friction 0.674 h.p. or	22,242
In cooling water $2.09 \times 106.9 \times 772$ or . .	172,500
Leaving for exhaust, waste, and radiation . .	533,921

Or reducing these numbers to percentages of the heat of combustion:

Per cent. given by brake, ...	18.12
" lost in engine friction, ...	2.50
Per cent. accounted for in indicator-diagram,	20.62
" given to jacket-water, ...	19.37
" lost in exhaust, etc., ...	60.00

Comparing these figures with those obtained previously, it will be found that (1) the efficiency in this case is about 3 per cent. or 4 per cent. greater = 3 or 4 per cent. more heat is converted into work in the cylinder; (2) less heat is given to the jacket-water and more is carried into the exhaust; that the gases retain the heat instead of its being given to the jacket-water and this increases the efficiency of the expansion.

Taking Card No. 10, we get the following values of the pressures and volumes in the cylinder:

	Pressure absolute	Volumes in cubic ft.
Beginning of compression, ...	14.7	0.1888
End of "	54.7	0.0738
Beginning of expansion, ...	167.7	0.0802
End of "	28.7	0.3102

Hence, since these curves are of the form

$$pv^n = \text{constant},$$

we get for the values of n—

For compression curve:

$$\frac{54.7}{14.7} = \left(\frac{0.1888}{0.0738}\right)^n;$$

$$n = 1.399.$$

For the expansion curve:

$$\frac{167.7}{28.7} = \left(\frac{0.3102}{0.0802}\right)^n$$

$$n = 1.305.$$

The expansion curve lies between an adiabatic and an isothermal, being nearer to an adiabatic. Hence the loss of heat to the jacket must be rather less than the heat developed during expansion. This is conformable to the fact observed of the rather low percentage of heat given to the jacket.

The following are the reported results of a similar trial made for the British Society of Arts.*

TRIAL OF AN ATKINSON GAS-ENGINE.

1	Date	Sept. 21	Sept. 22	Sept. 22
2	Trial	A	B	C
3	Duration	6 hours	3 hours	½ hour
4	Power	full	half	empty
5	Revolutions per minute	131.1	129.6	131.9
6	Explosions " "	121.6	69.1	23.8
7	Mean initial pressure	166.0	166.5	145.5
8	Mean effective pressure	46.07	47.60	48.59
9	Indicated H. P.	11.15	6.59	2.3
10	Brake-load net	130.5	66.0	....
11	Brake H. P.	9.48	4.74	....
12	Mechanical efficiency	0.850	0.719	....
13	Gas per hour, main	209.8	127.1	47.2
14	" " " ignition	4.5	5.9	....
15	" " " total	214.3	133.0	....
16	Gas per indicated H. P. per hour, main	18.82	19.29	20.50
17	" " " " " total.	19.22	20.18	....
18	Gas per brake H. P. per hour, main...	22.14	26.80	....
19	" " " " " " total....	22.61	28.10	....
20	Water per hour	680 lbs.	260 lbs.	....
21	Rise of temperature	52.2°	67.8°	....
22	H. P. in driving engine	1.67	1.85	2.3
23	Mean pressure during working stroke, equivalent to work done in pumping strokes, about	1.0	....	....
24	Corresponding indicated H. P.	0.26	....	....

* Journal, Feb. 15, 1889. p. 220.

The graphical record is as below:

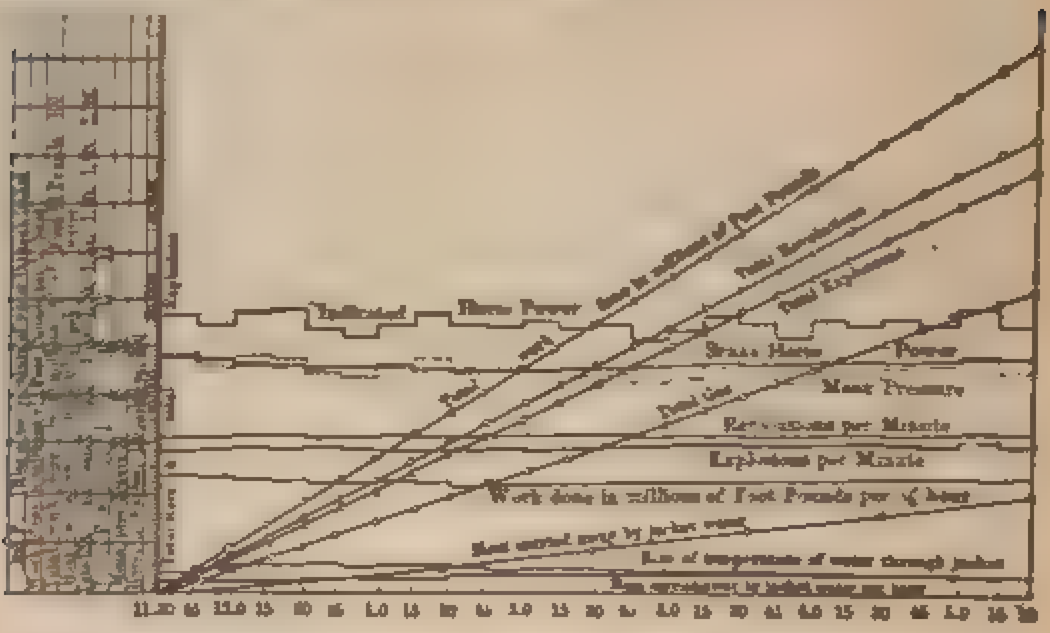


Fig. 132.—Gas-engine Trial.

From the log is obtained the following:

DISTRIBUTION OF ENERGY.

Heat turned into work as shown by indicator-diagrams,	22.8
Heat rejected in jacket-water, ...	27.0
Heat rejected in exhaust, lost by imperfect combustion, and otherwise accounted for, ...	50.2
	100.0

The actual expenditure of heat was at the rate of 11,250 thermal units per indicated h. p per hour, which corresponds to the absolute efficiency of 22.8 per cent. just given. It is very interesting to notice that the heat expenditure per indicated h. p. per hour is little more than half that of the steam-engine, a difference due, of course, to the greater range of temperature within which the engine works.

The efficiency of the engine, as compared with a perfect engine working between the same limits of temperature, and receiving the same amount of heat, is 28.2 per cent. The limits of temperature being assumed, the ideal mean diagram would give the following:

HEAT DISTRIBUTION.

	Foot-pounds per Explosion.	Percentage.
Calorific value of the gas used per explosion. 0.00[illegible] × [illegible] × 772	13,2[illegible]	1[illegible]
Heat turned into work ....!	[illegible]	[illegible]
Heat rejected in jacket-water........................	[illegible]	[illegible]
Heat rejected in exhaust...........................	[illegible]	[illegible]
Heat unaccounted for...............................	1,2[illegible]	[illegible]
	13,2[illegible]	100.0

There seems little doubt that the largeness of the percentage unaccounted for is due to the fact that combustion was not completed.

The same series of trials included an Otto engine built by the Crossley Works, which gave the following data:

TRIAL OF OTTO GAS-ENGINE.

1	Date	Sept. 19	Sept. 20	Sept. 20	Sept. 27
2	Trial....................................	A	B	C	D
3	Duration................................	6 hours	3 hours	¼ hour	¼ hour
4	Power	Full	Half	Empty	With and without counter shaft.
5	Revolutions per minute..................	160.1	158.8	161.0	[illegible] & 104.8
6	Explosions per minute...................	78.4	41.1	10.2	[illegible] & [illegible]
7	Mean initial pressure....	196.9	196.2	148.0	.. ...
8	Mean effective pressure........... ...	67.9	73.4	66.7	72.[illegible] & 74.1
9	Indicated H. P.................	17.12	9.73	2.19	4.40 & [illegible]
10	Brake-load net........................	177.4	89.9	....	....
11	Brake H. P.	14.74	7.41	....	....
12	Mechanical efficiency	0.861	0.762	....	....
13	Gas per hour, main	351.8	202.6	49.0	....
14	" " " ignition	3.5	3.2	....	....
15	" " " total	355.3	205.8	....	....
16	Gas per indicated H. P. per hour, main	20.55	20.8	22.38	....
17	" " " " " " total.	20.76	21.2	....	....
18	Gas per brake H. P. per hour, main....	23.87	27.34	....	....
19	" " " " " " total....	24.10	27 77	....	... .
20	" " net H. P. available for electric lighting per hour, after allowing for counter-shaft, as per trial D, main..	27.4	36.8	....	....
21	Water per hour	713 lbs.	480 lbs.	....	... ..
22	Rise of temperature...................	128.0°	102.3°	....	. . ....
23	H. P. in driving engine...............	2.38	2.31	2.19	2.[illegible]
24	Mean pressure during working stroke, equivalent to work done in pumping strokes, about.	2.19	....	....	....
25	Corresponding indicated H. P.	0.55	....	...	. .....

HEAT ACCOUNT.

	Foot pounds per Explosion.	Percentages
Calorific value of the gas per explosion 0.00223 × 19800 × 772	34,040	100
Heat turned into work..............................	7,515	22.1
Heat rejected in jacket-water......................	14,700	43.2
Heat rejected in exhaust...........................	12,100	35.5
		100.8

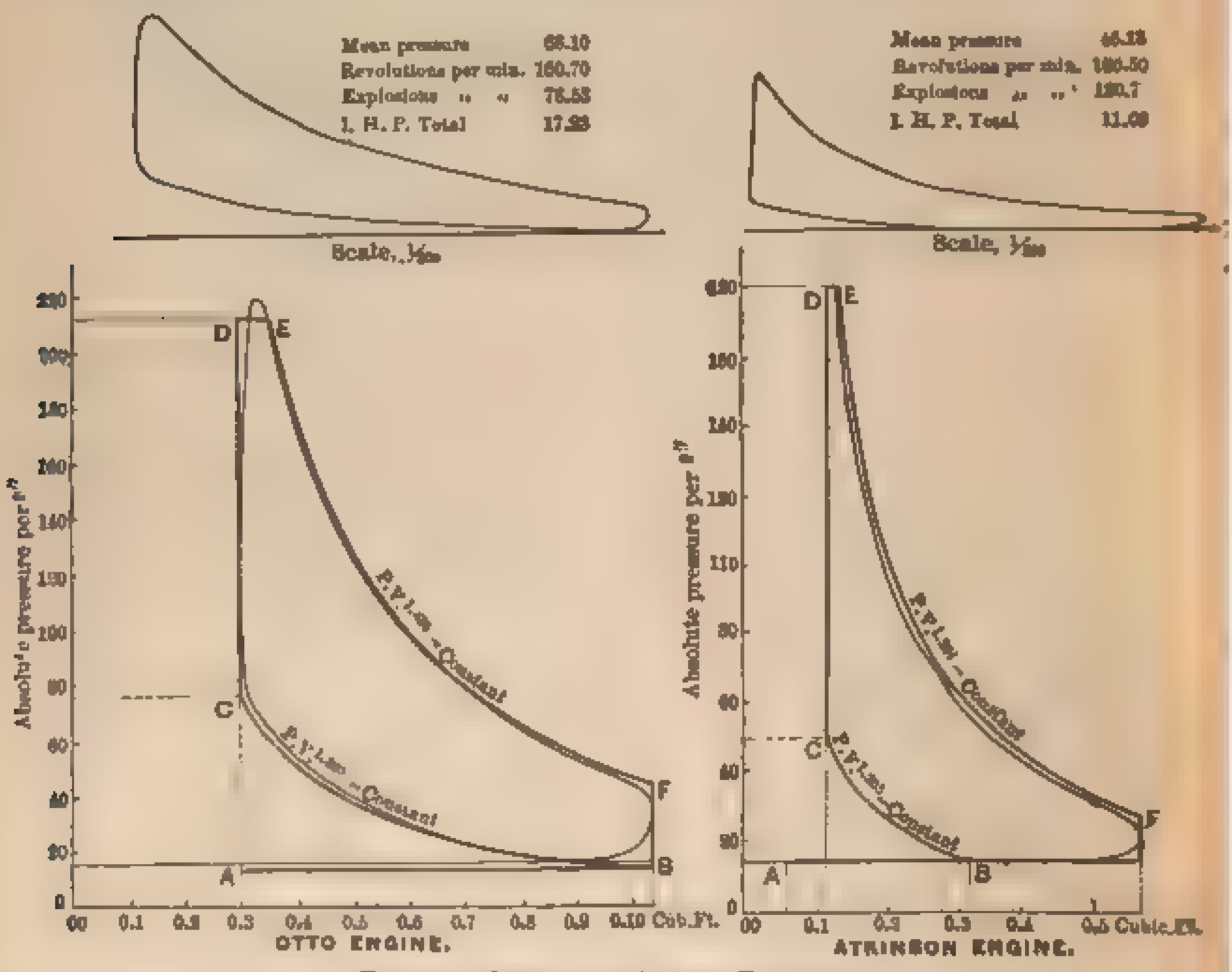


FIG. 133.—IDEAL AND ACTUAL DIAGRAMS.

Fig. 133 shows an indicator-diagram obtained from each of the two engines, and also the ideal diagram with which they are compared, both, in the latter case, being reduced to a common scale, and the one superposed over the other.*

* Journal Brit. Soc. Arts, Feb. 15, 1889.

105. Vapor and Binary-vapor Engine Trials have been seldom made in such manner as to yield results at all reliable. The trial of an engine designed by Du Trembley is reported by Rankine to have given the following results.*

The following are *means*, computed from results given in M. Gouin's report, on the performance of the steam and æther engines of the "Brésil":

	PRESSURE IN LBS. ON THE SQUARE INCH.		
	In boiler or evaporator.	Back pressure.	Mean effective.
Steam, ...	43.2	7.6	11.6
Æther, ...	31.2	5.3	7.1

Total M. E. P. reduced to the area of *one piston*, the areas and strokes of the pistons having been in this case the same, ... 18.7

It appears that the proportions of the power obtained in the cylinders, respectively, were:

$$\text{In the steam cylinder, } \frac{11.6}{18.7} = .62;$$

$$\text{In the æther cylinder, } \frac{7.1}{18.7} = .38.$$

The gain of power by the addition of the æther-engine is not quite so great as this shows; because, had the steam cylinder been used alone, the back pressure would have been in all probability about 4.6 instead of 7.6; so that the mean effective pressure in the steam cylinder would have been 14.6 instead of 11.6; and the proportion of the power of the steam-engine to that of the binary engine would have been

$$\frac{14.6}{18.7} = .77,$$

leaving

$$1.00 - .77 = .23$$

* Steam-engine, p. 447.

of the power of the binary engine, as the gain due to the æther-engine.

The consumption of fuel was either

$\left.\begin{array}{r}2.8\\ \text{or } 2.44\end{array}\right\}$ lbs. of coal per indicated horse-power per hour,

according as certain experiments made under peculiarly adverse circumstances were included or excluded. Rankine adds:

"The binary engine is not more economical than steam-engines designed with due regard to economy of fuel; but by the addition of an æther-engine, a wasteful steam-engine may be converted into an economical binary engine."

A binary-vapor engine, tested by Mr. Haswell, in which the auxiliary fluid was carbon disulphide, had the following construction: *

First, a horizontal cylindrical-fire tubular boiler.

Second, a tubular generator in form of a cylinder boiler, in which bisulphide of carbon (formula CS_2) is vaporized, having attached in the vapor space an ordinary perforated dry pipe, all inclosed in a shell having a diaphragm plate between the outer and inner shells at both sides and at one end, thus forming an upper and a lower chamber around it. The opposite end is inclosed with a deep disk or bonnet, forming a communication between the lower and upper series of tubes, for the proper circulation of the steam with which the CS_2 is vaporized.

Third, a horizontal non-condensing jacketed steam engine.

Fourth, conduit-pipe, steam-jacketed from the generator to the cylinder of the engine, the jacket of the conduit communicating with the jacket of the cylinder, and from thence the condensed steam led by a pipe to a steam-trap communicating with the feed-pump of the boiler.

Fifth, an automatic pressure-reducing valve, for controlling the admission of steam to the shell surrounding the generator, operated by the pressure assigned to the generator, holding the vapor-pressure uniform, by admitting more or less steam to the shell as the variation of the load on the engine may require.

* Trans. Am. Soc. C.E., 1887.

Sixth, a reducing valve for controlling the pressure in the jacket around the vapor-conduit and cylinder.

Seventh, a coil-heater through which the condensed CS_2 is forced back into the generator.

Eighth, a surface condenser.

Ninth, three small independent steam-pumps, and a connection to a water-main from which the water of condensation is obtained.

The generator is charged to a little over one half its capacity.

Steam is led by a pipe to and through the automatic regulating valve, where it is reduced in pressure and temperature; thence to the generator through a perforated pipe between the shells below; thence flowing around the lower half of the generator; thence through the lower tubes; thence through the upper series; and thence between the shells above, thus circulating through the entire generator, CS_2 taking up the heat; the steam is condensed and gravitates to the bottom of the outer shell; from thence to the boiler feed-pump, with the condensation from the jacket of the conduit and cylinder, delivered through the steam-trap, and from thence returned to the boiler.

Steam is also admitted through the reducing or regulating valve, to and through the jacket of the conduit to the jacket of the cylinder, where it is restricted to a reduced pressure, and as it is at a temperature due to this pressure, it is at a temperature in excess of that surrounding the generator, thus imparting an increased temperature to the vapor and superheating it.

The exhaust vapor from the cylinder passes around a coiled tube in the heater, thence through a surface condenser, from which it is drawn off by the second pump and delivered into an auxiliary condenser.

The liquid CS_2 gravitates from the auxiliary condenser to a reservoir. From thence it is drawn by the third pump and delivered through the coil in the heater, thence to the generator, where it is again vaporized.

A plant designed for the development of this design was tested.

The operation of the engine was continued five hours, which, as that period involved the cleaning of the fire, was held to afford time for a test.

The reported data are as below:

Pressure, steam—boiler,	75.8 pounds
" " shell,	15.3 "
" vapor—engine,	76 "
" " mean, by indicator,	31.35 "
Water evaporated,	5.71 cubic feet
Revolutions per minute,	100
Vacuum,	9.85 pounds
Coal consumed,	600 "
Horse-power indicated,	86.64

From which it appears that steam at a pressure of 75.8 pounds per square inch passed through the automatic regulating valve to the shell surrounding the generator at the reduced pressure of 15.3 pounds, due to a temperature of 250.4 degrees, produced a vapor in the generator of 76 pounds.

The consumption of coal was thus reported as 1.385 pounds per indicated horse-power per hour.

These results confirm the indications of thermodynamic science, that substantially as good work may be done with other vapors as with steam; but the steam-engine has actually given as good economical results as those here reported, and has many practical points of superiority. This trial was, however, too short to be taken as fully satisfactory, and the history of these devices, so far as known, does not seem to encourage an expectation of the displacement of the steam-engine by their introduction.

The data and results obtained by Mr. Barrus, by test of a Campbell *ammonia-engine* and boiler, as reported to the Campbell Engine Co., April 1887, were as follow:

Table No. 1.

Principal Dimensions of Boiler and Engine.

Boiler—One Horizontal Return Tubular, set in brick-work.

1	Diameter of shell..........................	42 in.
2	Length of shell..........................	10 ft.
3	Number of 2 in. tubes below level of liquid..	67
4	" " " " above " " "	68
5	Inside diameter of tubes..................	1.75 in.
6	Length of grate (net)....................	2.75 ft.
7	Width of grate	3.33 "
8	Width of metal and air-spaces in grate......	⅜ in.
9	Area of water-heating surface..............	369.3 sq. ft.
10	Area of steam-heating surface	318.8 "
11	Area of grate-surface......................	9.17 "
12	Collective area for draught through 67 tubes	1.12 "
13	Rated horse-power of boiler on basis of 15 sq. ft. of water-heating surface per h. p....	24.6 h. p.
14	Ratio of water-heating surface to grate-surface	40.3 to 1
15	Ratio of steam-heating surface to grate-surface..................................	33.6 to 1
16	Ratio of grate-surface to tube area.........	8.2 to 1
17	Diameter of smoke-stack..................	20. in.
18	Height of smoke-stack above grate.........	30. ft.

Engine—Porter-Allen Automatic Cut-off, Single Cylinder.

19	Diameter of cylinder	11.5 in.
20	Stroke of piston	20 "
21	Diameter of piston-rod	1¾ "
22	Clearance (assumed)......................	10 per cent.
23	Horse-power constant 1 lb. M. E. P. 1 rev. per minute............................	0.01037 h. p.
24	Diameter of steam-pipe....................	4. in.

TABLE No. 2.

Ammonia Tests of Whole Plant.

		March 8,	March 9,	April 16.
[illegible]	[illegible]	8	10	7.45
[illegible]	lbs.	1400	1400	1098
[illegible]	"		139	90
[illegible]	per cent.		9.9	8.2
[illegible]	lbs.	175	140	147.4
[illegible]	"	19.09	15.27	16.07
[illegible]	cu. ft.			372.8
[illegible]	cu. ft.	Approx. 100	95.5	86.6
[illegible]	Deg. F.		304	301
[illegible]	"			180.8
[illegible]	"		280	271
[illegible]	"		[illegible]	[illegible]
[illegible]	"	Approx. [illegible]	[illegible]	541
[illegible]	"		[illegible]	[illegible]
[illegible]	"			[illegible]

TABLE NO. 2. (Continued.)

Data and Results of Ammonia Tests of Whole Plant.

Date........	1887,	March 8,	March 9,	April 16.
16 Temp. of injection-water from river.....	Deg. F.			42.7
17 Temp. of same leaving the absorbers.......	"			104.2
18 Vacuum in feed-well...	inches,		11.5	11
19 Revolutions of engine per minute.........	revolu.	205.2	204.5	201.5
20 Mean effective pressure measured from indicator-diagrams......	lbs.	29.05	27.13	*26.99
21 Indicated horse-power developed by engine.	H. P.	61.80	57.53	54.00
22 Weather and outside temperature......			Fair Moderate	Fair Moderate
Averages of measurements of two sets of Sample Diagrams.				
23 Boiler-pressure above atmosphere.........	lbs.		96.5	88.2
24 Initial pressure above atmosphere..... ...	"		88.7	80.8
25 Cut-off pressure above atmosphere..... ...	"		67.9	61.5
26 Release pressure above atmosphere.........	"		9.6	9.1
27 Compression pressure above atmosphere...	"		1.7	0
28 Back-pressure at mid-stroke below atmosphere.............	"		0.7	1.5

* Corresponding to the normal load of 56.39 H. P.

TABLE NO. 2.

Data and Results of Ammonia Test

Date	1887, March		
1 Duration of test	hrs.	8	
Total Quantities.			
2 Weight of coal consumed	lbs.	14[illegible]	
3 Weight of ashes, clinkers, and refuse	"		.211
4 Percentage of ashes, etc.	per cent	5	.791
Hourly Quantities.			
5 Coal consumed per hour	lbs.		
6 Coal per hour per sq. ft. of grate	"	.307	.342
7 Quantity of injection-water used with normal load	cu.	2.433	2.729
[illegible]			
8 [illegible] pressure above atmosphere	[illegible]		23.760
9 [illegible] vapor in main [illegible] throttle			
10 [illegible] vapor in ex[illegible]			

[illegible] Experience, as illustrated [illegible] leads to such final conclu[illegible]

[illegible] and the philosophy of heat[illegible] the limit of possible advance [illegible] now so nearly approached [illegible] expected to be both slow and toil[illegible]

[illegible] further improvement upon [illegible] is probably small. [illegible] to reduce it are in[illegible]

than steam, no other vapor has yet been found to give an economical performance in heat-transformation exceeding, or even equalling, that obtained with the best steam-engines.

(4) That the gas-engine, with its higher range of temperature variation, is the most promising competitor: but that it is not yet possible to judge whether the increase of temperature-range to be expected with the steam-engine, or the increased pressure-range of the gas-engine, with the possibilities of waste-reduction seen to be within reach, is likely to give the one or the other final superiority for small powers.

The outlook in the case of steam, used as a working fluid, is well shown by the illustration given on the next page, from a paper by Mr. Parker.* The diagram shows the method of expansion of steam at an absolute pressure of 140 pounds per square inch (9½ atmos.): (*a*) when kept dry and saturated; (*b*) when expanding adiabatically; (*c*) and as actually worked in the steam-cylinders of the S. S. "Aberdeen," designed by Mr. Kirk for the China trade, an example of exceptional economy.† It is seen that the actual expansion line was bounded very closely by the adiabatic line, thus showing the internal condensation to be variable in a manner similar to that in a non-conducting cylinder. The jacket-wastes, however, amounting to about 4 per cent., must be added to the quantity of steam here shown. The table accompanying the diagram exhibits the computed adiabatic condensation for the full range of expansion, varying from 0, at the start, to 14.7 per cent. at the end. For ordinary engines, at lower pressures, it would become, on this scale, about 4 per cent. for steam at 70 pounds absolute pressure, 6 at 47, 7½ at 35, 10 at 23, 11 at 18, 12 at 14, and 14 per cent. at 8½; or at ratios of expansion, respectively, of 2, 3, 4, 6, 8, 10, and 15.

The wastes due to action of valves, the loss in passages, and to maladjustment of the several parts of the system to each

* Economy of Compound Engines; Trans. Brit Inst. N. A., 1882.

† Engines: 30, 45, and 70 inch cylinders, the first unjacketed; 4½ feet stroke of piston. Steam 125 lbs. by gauge, in boiler; 125, 50, and 15 lbs. in jackets. H.P. 1800; fuel per h. p. per hr., 1.28 lbs.

... seen, as in other cases already presented, in the variation of the ideal diagrams from their respective portions of the ... These wastes may be reduced somewhat by improved design and construction; but, on the whole, they in-

Full line shows relative pressures and volumes, steam being dry and saturated.
Dotted line shows relative pressures and volumes, steam expanding adiabatically.

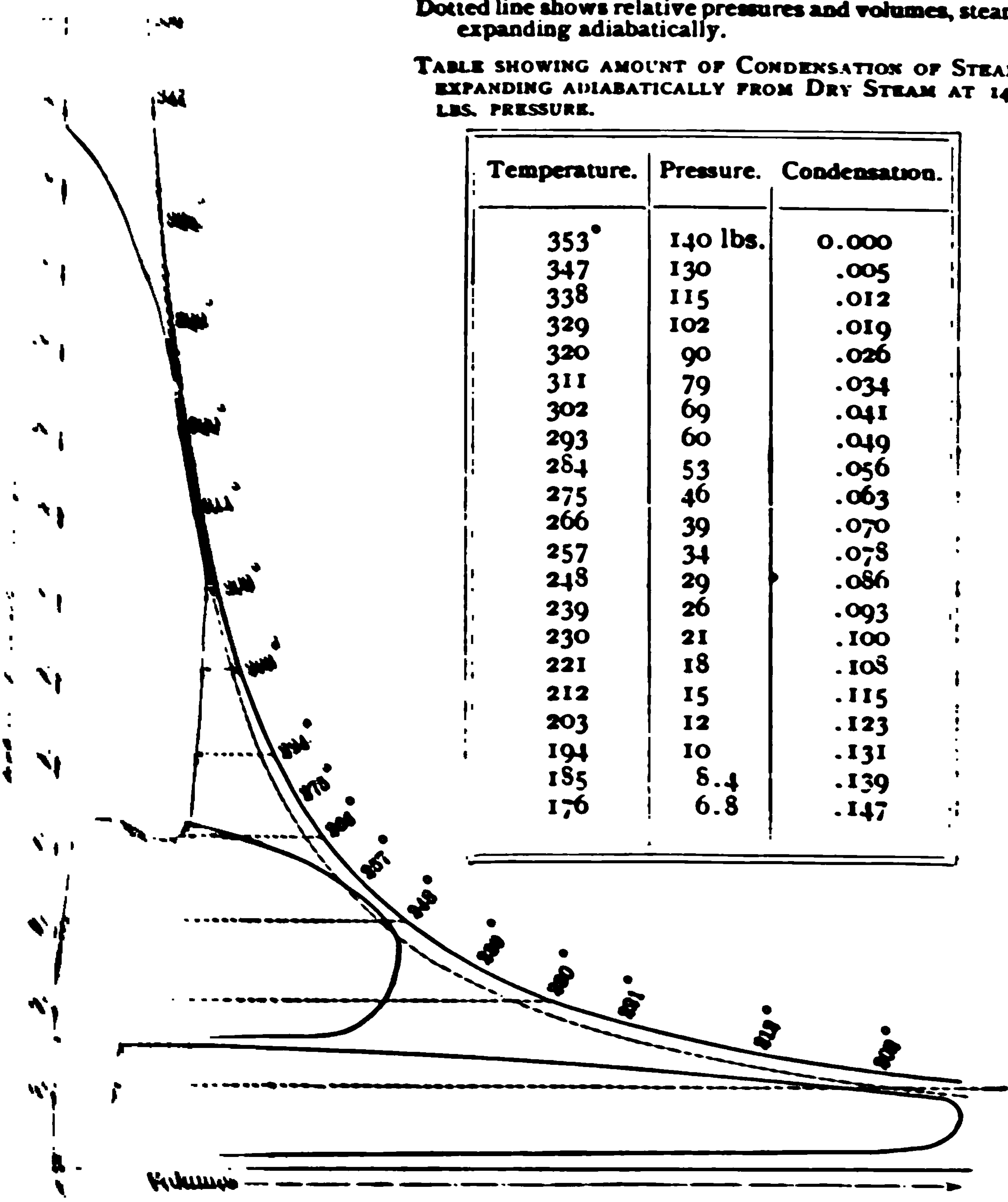


TABLE SHOWING AMOUNT OF CONDENSATION OF STEAM EXPANDING ADIABATICALLY FROM DRY STEAM AT 140 LBS. PRESSURE.

Temperature.	Pressure.	Condensation.
353°	140 lbs.	0.000
347	130	.005
338	115	.012
329	102	.019
320	90	.026
311	79	.034
302	69	.041
293	60	.049
284	53	.056
275	46	.063
266	39	.070
257	34	.078
248	29	.086
239	26	.093
230	21	.100
221	18	.108
212	15	.115
203	12	.123
194	10	.131
185	8.4	.139
176	6.8	.147

FIG. 134.—ECONOMY OF STEAM.

crease with higher pressures and greater expansion, and thus aggravate the difficulties of securing higher economy. All these points being considered, the gain by still higher pressure is seen to be comparatively small, as is evidenced in part by

the small and contracting area of the diagram at its top, as compared with its area nearer the base, corresponding to lower initial pressures.

It would seem probable that future progress in the economical operation of heat-engines must be slow in any one of its several possible directions, and the aggregate only likely to exhibit gain in any considerable degree by practically suppressing internal wastes and then greatly increasing the range of adiabatic heat and temperature variation.

APPENDIX.

TABLES.

I.

NUMERICAL CONSTANTS.

n	$n\pi$	$n^2\frac{\pi}{4}$	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$
1.0	3.142	0.7854	1.000	1.000	1.0000	1.0000
1.1	3.456	0.9503	1.210	1.331	1.0488	1.0323
1.2	3.770	1.1310	1.440	1.728	1.0955	1.0627
1.3	4.084	1.3273	1.690	2.197	1.1402	1.0914
1.4	4.398	1.5394	1.960	2.744	1.1832	1.1187
1.5	4.712	1.7672	2.250	3.375	1.2247	1.1447
1.6	5.027	2.0106	2.560	4.096	1.2649	1.1696
1.7	5.341	2.2698	2.890	4.913	1.3038	1.1935
1.8	5.655	2.5447	3.240	5.832	1.3416	1.2164
1.9	5.969	2.8353	3.610	6.859	1.3784	1.2386
2.0	6.283	3.1416	4.000	8.000	1.4142	1.2599
2.1	6.597	3.4636	4.410	9.261	1.4491	1.2806
2.2	6.912	3.8013	4.840	10.648	1.4832	1.3006
2.3	7.226	4.1548	5.290	12.167	1.5166	1.3200
2.4	7.540	4.5239	5.760	13.824	1.5492	1.3389
2.5	7.854	4.9087	6.250	15.625	1.5811	1.3572
2.6	8.168	5.3093	6.760	17.576	1.6125	1.3751
2.7	8.482	5.7256	7.290	19.683	1.6432	1.3925
2.8	8.797	6.1575	7.840	21.952	1.6733	1.4095
2.9	9.111	6.6052	8.410	24.389	1.7029	1.4260
3.0	9.425	7.0686	9.00	27.000	1.7321	1.4422
3.1	9.739	7.5477	9.61	29.791	1.7607	1.4581
3.2	10.053	8.0425	10.24	32.768	1.7889	1.4736
3.3	10.367	8.5530	10.89	35.937	1.8166	1.4888
3.4	10.681	9.0792	11.56	39.304	1.8439	1.5037
3.5	10.996	9.6211	12.25	42.875	1.8708	1.5183
3.6	11.310	10.179	12.96	46.656	1.8974	1.5326
3.7	11.624	10.752	13.69	50.653	1.9235	1.5467
3.8	11.938	11.341	14.44	54.872	1.9494	1.5605
3.9	12.252	11.946	15.21	59.319	1.9748	1.5741
4.0	12.566	12.566	16.00	64.000	2.0000	1.5874
4.1	12.881	13.203	16.81	68.921	2.0249	1.6005
4.2	13.195	13.854	17.64	74.088	2.0494	1.6134
4.3	13.509	14.522	18.49	79.507	2.0736	1.6261
4.4	13.823	15.205	19.36	85.184	2.0976	1.6386
4.5	14.137	15.904	20.25	91.125	2.1213	1.6510
4.6	14.451	16.619	21.16	97.336	2.1448	1.6631
4.7	14.765	17.349	22.09	103.823	2.1680	1.6751

CONSTANTS—*Continued.*

n	$n\pi$	$n^2\frac{\pi}{4}$	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$
4.8	15.080	18.096	23.04	110.592	2.1909	1.6869
4.9	15.394	18.857	24.01	117.649	2.2136	1.6985
5.0	15.708	19.635	25.00	125.000	2.2361	1.7100
5.1	16.022	20.428	26.01	132.651	2.2583	1.7213
5.2	16.336	21.237	27.04	140.608	2.2804	1.7325
5.3	16.650	22.062	28.09	148.877	2.3022	1.7435
5.4	16.965	22.902	29.16	157.464	2.3238	1.7544
5.5	17.279	23.758	30.25	166.375	2.3452	1.7652
5.6	17.593	24.630	31.36	175.616	2.3664	1.7758
5.7	17.907	25.518	32.49	185.193	2.3875	1.7863
5.8	18.221	26.421	33.64	195.112	2.4083	1.7967
5.9	18.535	27.340	34.81	205.379	2.4290	1.8070
6.0	18.850	28.274	36.00	216.000	2.4495	1.8171
6.1	19.164	29.225	37.21	226.981	2.4698	1.8272
6.2	19.478	30.191	38.44	238.328	2.4900	1.8371
6.3	19.792	31.173	39.69	250.047	2.5100	1.8469
6.4	20.106	32.170	40.96	262.144	2.5298	1.8566
6.5	20.420	33.183	42.25	274.625	2.5495	1.8663
6.6	20.735	34.212	43.56	287.496	2.5691	1.8758
6.7	21.049	35.257	44.89	300.763	2.5884	1.8852
6.8	21.363	36.317	46.24	314.432	2.6077	1.8945
6.9	21.677	37.393	47.61	328.509	2.6268	1.9038
7.0	21.991	38.485	49.00	343.000	2.6458	1.9129
7.1	22.305	39.592	50.41	357.911	2.6646	1.9220
7.2	22.619	40.715	51.84	373.248	2.6833	1.9310
7.3	22.934	41.854	53.29	389.017	2.7019	1.9399
7.4	23.248	43.008	54.76	405.224	2.7203	1.9487
7.5	23.562	44.179	56.25	421.875	2.7386	1.9574
7.6	23.876	45.365	57.76	438.976	2.7568	1.9661
7.7	24.190	46.566	59.29	456.533	2.7749	1.9747
7.8	24.504	47.784	60.84	474.552	2.7929	1.9832
7.9	24.819	49.017	62.41	493.039	2.8107	1.9916
8.0	25.133	50.266	64.00	512.000	2.8284	2.0000
8.1	25.447	51.530	65.61	531.441	2.8461	2.0083
8.2	25.761	52.810	67.24	551.468	2.8636	2.0165
8.3	26.075	54.106	68.89	571.787	2.8810	2.0247
8.4	26.389	55.418	70.56	592.704	2.8983	2.0328
8.5	26.704	56.745	72.25	614.125	2.9155	2.0408
8.6	27.018	58.088	73.96	636.056	2.9326	2.0488
8.7	27.332	59.447	75.69	658.503	2.9496	2.0567
8.8	27.646	60.821	77.44	681.473	2.9665	2.0646
8.9	27.960	62.211	79.21	704.969	2.9833	2.0724

CONSTANTS—*Continued.*

n	$n\pi$	$n^2\frac{\pi}{4}$	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$
9.0	28.274	63.617	81.00	729.000	3.0000	2.0801
9.1	28.588	65.039	82.81	753.571	3.0166	2.0878
9.2	28.903	66.476	84.64	778.688	3.0332	2.0954
9.3	29.217	67.929	86.49	804.357	3.0496	2.1029
9.4	29.531	69.398	88.36	830.584	3.0659	2.1105
9.5	29.845	70.882	90.25	857.375	3.0822	2.1179
9.6	30.159	72.382	92.16	884.736	3.0984	2.1253
9.7	30.473	73.898	94.09	912.673	3.1145	2.1327
9.8	30.788	75.430	96.04	941.192	3.1305	2.1400
9.9	31.102	76.977	98.01	970.299	3.1464	2.1472
10.0	31.416	78.540	100.00	1000.000	3.1623	2.1544
10.1	31.730	80.119	102.01	1030.301	3.1780	2.1616
10.2	32.044	81.713	104.04	1061.208	3.1937	2.1687
10.3	32.358	83.323	106.09	1092.727	3.2094	2.1757
10.4	32.673	84.949	108.16	1124.863	3.2249	2.1828
10.5	32.987	86.590	110.25	1157.625	3.2404	2.1897
10.6	33.301	88.247	112.36	1191.016	3.2558	2.1967
10.7	33.615	89.920	114.49	1225.043	3.2711	2.2036
10.8	33.929	91.609	116.64	1259.712	3.2863	2.2104
10.9	34.243	93.313	118.81	1295.029	3.3015	2.2172
11.0	34.558	95.033	121.00	1331.000	3.3166	2.2239
11.1	34.872	96.769	123.21	1367.631	3.3317	2.2307
11.2	35.186	98.520	125.44	1404.928	3.3466	2.2374
11.3	35.500	100.29	127.69	1442.897	3.3615	2.2441
11.4	35.814	102.07	129.96	1481.544	3.3764	2.2506
11.5	36.128	103.87	132.25	1520.875	3.3912	2.2572
11.6	36.442	105.68	134.56	1560.896	3.4059	2.2637
11.7	36.757	107.51	136.89	1601.613	3.4205	2.2702
11.8	37.071	109.36	139.24	1643.032	3.4351	2.2766
11.9	37.385	111.22	141.61	1685.159	3.4496	2.2831
12.0	37.699	113.10	144.00	1728.000	3.4641	2.2894
12.1	38.013	114.99	146.41	1771.561	3.4785	2.2957
12.2	38.327	116.90	148.84	1815.848	3.4928	2.3021
12.3	38.642	118.82	151.29	1860.867	3.5071	2.3084
12.4	38.956	120.76	153.76	1906.624	3.5214	2.3146
12.5	39.270	122.72	156.25	1953.125	3.5355	2.3208
12.6	39.584	124.69	158.76	2000.376	3.5496	2.3270
12.7	39.898	126.68	161.29	2048.383	3.5637	2.3331
12.8	40.212	128.68	163.84	2097.152	3.5777	2.3392
12.9	40.527	130.70	166.41	2146.689	3.5917	2.3453
13.0	40.841	132.73	169.00	2197.000	3.6056	2.3513
13.1	41.155	134.78	171.61	2248.091	3.6194	2.3573
13.2	41.469	136.85	174.24	2299.968	3.6332	2.3633

CONSTANTS—*Continued.*

n	$n\pi$	$n^2\frac{\pi}{4}$	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$
13 3	41.783	138.93	176.89	2352.637	3.6469	2.3693
13.4	42.097	141.03	179.56	2406.104	3.6606	2.3752
13.5	42.412	143.14	182.25	2460.375	3.6742	2.3811
13.6	42.726	145.27	184.96	2515.456	3.6878	2.3870
13.7	43.040	147.41	187.69	2571.353	3.7013	2.3928
13.8	43.354	149.57	190.44	2628.072	3.7148	2.3986
13.9	43.668	151.75	193.21	2685.619	3.7283	2.4044
14.0	43.982	153.94	196.00	2744.000	3.7417	2.4101
14.1	44.296	156.15	198.81	2803.221	3.7550	2.4159
14.2	44.611	158.37	201.64	2863.288	3.7683	2.4216
14.3	44.925	160.61	204.49	2924.207	3.7815	2.4272
14.4	45.239	162.86	207.36	2985.984	3.7947	2.4329
14.5	45.553	165.13	210.25	3048.625	3.8079	2.4385
14.6	45.867	167.42	213.16	3112.136	3.8210	2.4441
14.7	46.181	169.72	216.09	3176.523	3.8341	2.4497
14.8	46.496	172.03	219.04	3241.792	3.8471	2.4552
14.9	46.810	174.37	222.01	3307.949	3.8600	2.4607
15.0	47.124	176.72	225.00	3375.000	3.8730	2.4662
15.1	47.438	179.08	228.01	3442.951	3.8859	2.4717
15.2	47.752	181.46	231.04	3511.808	3.8987	2.4772
15.3	48.066	183.85	234.09	3581.577	3.9115	2.4825
15.4	48.381	186.27	237.16	3652.264	3.9243	2.4879
15.5	48.695	188.69	240 25	3723.875	3.9370	2.4933
15.6	49.009	191.13	243.36	3796.416	3.9497	2.4986
15.7	49.323	193.59	246.49	3869.893	3.9623	2.5039
15.8	49.637	196.07	249.64	3944.312	3.9749	2.5092
15.9	49.951	198.56	252.81	4019.679	3.9875	2.5146
16.0	50.265	201.06	256.00	4096.000	4.0000	2.5198
16.1	50.580	203.58	259.21	4173.281	4.0125	2.5251
16.2	50.894	206.12	262.44	4251.528	4.0249	2.5303
16.3	51.208	208.67	265.69	4330.747	4.0373	2.5355
16.4	51.522	211.24	268.96	4410.944	4.0497	2.5406
16.5	51.836	213.83	272.25	4492.125	4.0620	2.5458
16.6	52 150	216.42	275.56	4574.296	4.0743	2.5509
16.7	52.465	219.04	278.89	4657.463	4.0866	2.5561
16.8	52.779	221.67	282.24	4741.632	4.0988	2.5612
16.9	53.093	224.32	285.61	4826.809	4.1110	2.5663
17.0	53.407	226 98	289.00	4913.000	4.1231	2.5713
17.1	53.721	229.66	292.41	5000.211	4.1352	2.5763
17.2	54.035	132.35	295.84	5088.448	4.1473	2.5813
17.3	54.350	235.06	299.29	5177.717	4.1593	2.5863
17.4	54.664	237.79	302.76	5268.024	4.1713	2.5913

Constants—*Continued.*

n	πn	$n^2 \frac{\pi}{4}$	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$
17.5	54.978	240.53	306.25	5359.375	4.1833	2.5962
17.6	55.292	243.29	309.76	5451.776	4.1952	2.6012
17.7	55.606	246.06	313.29	5545.233	4.2071	2.6061
17.8	55.920	248.85	316.84	5639.752	4.2190	2.6110
17.9	56.235	251.65	320.41	5735.339	4.2308	2.6158
18.0	56.549	254.47	324.00	5832.000	4.2426	2.6207
18.1	56.863	257.30	327.61	5929.741	4.2544	2.6256
18.2	57.177	260.16	331.24	6028.568	4.2661	2.6304
18.3	57.491	263.02	334.89	6128.487	4.2778	2.6352
18.4	57.805	265.90	338.56	6229.504	4.2895	2.6401
18.5	58.119	268.80	342.25	6331.625	4.3012	2.6448
18.6	58.434	271.72	345.96	6434.856	4.3128	2.6495
18.7	58.748	274.65	349.69	6539.203	4.3243	2.6543
18.8	59.062	277.59	353.44	6644.672	4.3359	2.6590
18.9	59.376	280.55	357.21	6751.269	4.3474	2.6637
19.0	59.690	283.53	361.00	6859.000	4.3589	2 6684
19.1	60.004	286.52	364.81	6967.871	4.3703	2.6731
19.2	60.319	289.53	368.64	7077.888	4.3818	2.6777
19.3	60.633	292.55	372.49	7189.057	4.3932	2.6824
19.4	60.947	295.59	376.36	7301.384	4.4045	2.6869
19.5	61.261	298.65	380.25	7414.875	4.4159	2.6916
19.6	61.575	301.72	384.16	7529.536	4.4272	2.6962
19.7	61.889	304.81	388.09	7645.373	4.4385	2.7008
19.8	62.204	307.91	392.04	7762.392	4.4497	2.7053
19.9	62.518	311.03	396.01	7880.599	4.4609	2.7098
20.0	62.832	314.16	400.00	8000.000	4.4721	2.7144
20.1	63.146	317.31	404.01	8120.601	4.4833	2.7189
20.2	63.460	320.47	408.04	8242.408	4.4944	2.7234
20.3	63.774	323.66	412.09	8365.427	4.5055	2.7279
20.4	64.088	326 85	416.16	8489.664	4.5166	2.7324
20.5	64.403	330.06	420.25	8615.125	4.5277	2.7368
20.6	64.717	333.29	424.36	8741.816	4.5387	2.7413
20.7	65.031	336.54	428.49	8869.743	4.5497	2.7457
20.8	65.345	339.80	432.64	8989.912	4.5607	2.7502
20.9	05.659	343.07	436.81	9129.329	4.5716	2.7545
21.0	65.973	346.36	441.00	9261.000	4.5826	2.7589
21.1	66.288	349.67	445.21	9393.931	4.5935	2.7633
21.2	66.602	352.99	449.44	9528.128	4.6043	2.7676
21.3	66.916	356.33	453.69	9663.597	4.6152	2.7720
21.4	67.230	359.68	457.96	9800.344	4.6260	2.7763
21.5	67.544	363.05	462.25	9938.375	4.6368	2.7806
21.6	67.858	366.44	466.56	10077.696	4.6476	2.7849
21.7	68.173	369.84	470.89	10218.313	4.6583	2.7893

CONSTANTS—*Continued.*

n	$n\pi$	$n^2\frac{\pi}{4}$	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$
21.8	68.487	373.25	475.24	10360.232	4.6690	2.7935
21.9	68.801	376.69	479.61	10503.459	4.6797	2.7978
22.0	69.115	380.13	484.00	10648.000	4.6904	2.8021
22.1	69.429	383.60	488.41	10793.861	4.7011	2.8063
22.2	69.743	387.08	492.84	10941.048	4.7117	2.8105
22.3	70.058	390.57	497.29	11089.567	4.7223	2.8147
22.4	70.372	394.08	501.76	11239.424	4.7329	2.8189
22.5	70.686	397.61	506.25	11390.625	4.7434	2.8231
22.6	71.000	401.15	510.76	11543.176	4.7539	2.8273
22.7	71.314	404.71	515.29	11697.083	4.7644	2.8314
22.8	71.268	408.28	519.84	11852.352	4.7749	2.8356
22.9	71.942	411.87	524.41	12008.989	4.7854	2.8397
23.0	72.257	415.48	529.00	12167.000	4.7958	2.8438
23.1	72.571	419.10	533.61	12326.391	4.8062	2.8479
23.2	72.885	422.73	538.24	12487.168	4.8166	2.8521
23.3	73.199	426.39	542.89	12649.337	4.8270	2.8562
23.4	73.513	430.05	547.56	12812.904	4.8373	2.8603
23.5	73.827	433.74	552.25	12977.875	4.8477	2.8643
23.6	74.142	437.44	556.96	13144.256	4.8580	2.8684
23.7	74.456	441.15	561.69	13312.053	4.8683	2.8724
23.8	74.770	444.88	566.44	13481.272	4.8785	2.8765
23.9	75.084	448.63	571.21	13651.919	4.8888	2.8805
24.0	75.398	452.39	576.00	13824.000	4.8990	2.8845
24.1	75.712	456.17	580.81	13997.521	4.9092	2.8885
24.2	76.027	459.96	585.64	14172.488	4.9193	2.8925
24.3	76.341	463.77	590.49	14348.907	4.9295	2.8965
24.4	76.655	467.60	595.36	14526.784	4.9396	2.9004
24.5	76.969	471.44	600.25	14706.125	4.9497	2.9044
24.6	77.283	475.29	605.16	14886.936	4.9598	2.9083
24.7	77.597	479.16	610.09	15069.223	4.9699	2.9123
24.8	77.911	483.05	615.04	15252.992	4.9799	2.9162
24.9	78.226	486.96	620.01	15438.249	4.9899	2.9201
25.0	78.540	490.87	625.00	15625.000	5.0000	2.9241
25.1	78.854	494.81	630.01	15813.251	5.0099	2.9279
25.2	79.168	498.76	635.04	16003.008	5.0199	2.9318
25.3	79.482	502.73	640.09	16194.277	5.0299	2.9356
25.4	79.796	506.71	645.16	16387.064	5.0398	2.9395
25.5	80.111	510.71	650.25	16581.375	5.0497	2.9434
25.6	80.425	514.72	655.36	16777.216	5.0596	2.9472
25.7	80.739	518.75	660.49	16974.593	5.0695	2.9510
25.8	81.053	522.79	665.64	17173.512	5.0793	2.9549
25.9	81.367	526.85	670.81	17373.979	5.0892	2.9586

CONSTANTS—*Continued.*

n	$n\pi$	$n^2\frac{\pi}{4}$	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$
26.0	81.681	530.93	676.00	17576.000	5.0990	2.9624
26.1	81.996	535.02	681.21	17779.581	5.1088	2.9662
26.2	82.310	539.13	686.44	17984.728	5.1185	2.9701
26.3	82.624	543.25	691.69	18191.447	5.1283	2.9738
26.4	82.938	547.39	696.96	18399.744	5.1380	2.9776
26.5	83.252	551.55	702.25	18609.625	5.1478	2.9814
26.6	83.566	555.72	707.56	18821.096	5.1575	2.9851
26.7	83.881	559.90	712.89	19034.163	5.1672	2.9888
26.8	84.195	564.10	718.24	19248.832	5.1768	2.9926
26.9	84.509	568.32	723.61	19465.109	5.1865	2.9963
27.0	84.823	572.56	729.00	19683.000	5.1962	3.0000
27.1	85.137	576.80	734.41	19902.511	5.2057	3.0037
27.2	85.451	581.07	739.84	20123.648	5.2153	3.0074
27.3	85.765	585.35	745.29	20346.417	5.2249	3.0111
27.4	86.080	589.65	750.76	20570.824	5.2345	3.0147
27.5	86.394	593.96	756.25	20796.875	5.2440	3.0184
27.6	86.708	598.29	761.76	21024.576	5.2535	3.0221
27.7	87.022	602.63	767.29	21253.933	5.2630	3.0257
27.8	87.336	606.99	772.84	21484.952	5.2725	3.0293
27.9	87.650	611.36	778.41	21717.639	5.2820	3.0330
28.0	87.965	615.75	784.00	21952.000	5.2915	3.0366
28.1	88.279	620.16	789.61	22188.041	5.3009	3.0402
28.2	88.593	624.58	795.24	22425.768	5.3103	3.0438
28.3	88.907	629.02	800.89	22665.187	5.3197	3.0474
28.4	89.221	633.47	806.56	22906.304	5.3291	3.0510
28.5	89.535	637.94	812.25	23149.125	5.3385	3.0546
28.6	89.850	642.42	817.96	23393.656	5.3478	3.0581
28.7	90 164	646.93	823.69	23639.903	5.3572	3.0617
28.8	90.478	651.44	829.44	23887.872	5.3665	3.0652
28.9	90.792	655.97	835.21	24137.569	5.3758	3.0688
29.0	91.106	660.52	841.00	24389.000	5.3852	3.0723
29.1	91.420	665.08	846.81	24642.171	5.3944	3.0758
29.2	91.735	669.66	852.64	24897.088	5.4037	3.0794
29.3	92.049	674.26	858.49	25153.757	5.4129	3.0829
29.4	92.363	678.87	864.36	25412.184	5.4221	3.0864
29 5	92.677	683.49	870.25	25672.375	5.4313	3.0899
29.6	92.991	688.13	876.16	25934.336	5.4405	3.0934
29.7	93.305	692.79	882.09	26198.073	5.4497	3.0968
29.8	93.619	697.47	888.04	26463.592	5.4589	3.1003
29.9	93.934	702.15	894.01	26730.899	5.4680	3.1038
30.0	94.248	706.86	900.00	27000.000	5.4772	3.1072
30.1	94.562	711.58	906.01	27270.901	5.4863	3.1107
30.2	94.876	716.32	912.04	27543.608	5.4954	3.1141

CONSTANTS—*Continued.*

n	πn	$n^2\frac{\pi}{4}$	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$
30.3	95.190	721.07	918.09	27818.127	5.5045	3.1170
30.4	95.505	725.83	924.16	28094.464	5.5136	3.1210
30.5	95.819	730.62	930.25	28372.625	5.5226	3.1244
30.6	96.133	735.42	936.36	28652.616	5.5317	3.1278
30.7	96.447	740.23	942.49	28934.443	5.5407	3.1312
30.8	96.761	745.06	948.64	29218.112	5.5497	3.1346
30.9	97.075	749.91	954.81	29503.629	5.5587	3.1380
31.0	97.389	754.77	961.00	29791.000	5.5678	3.1414
31.1	97.704	759.65	967.21	30080.231	5.5767	3.1448
31.2	98.018	764.54	973.44	30371.328	5.5857	3.1481
31.3	98.332	769.45	979.69	30664.297	5.5946	3.1515
31.4	98.646	774.37	985.96	30959.144	5.6035	3.1548
31.5	98.960	779.31	992.25	31255.875	5.6124	3.1582
31.6	99.274	784.27	998.56	31554.496	5.6213	3.1615
31.7	99.588	789.24	1004.89	31855.013	5.6302	3.1648
31.8	99.903	794.23	1011.24	32157.432	5.6391	3.1681
31.9	100.22	799.23	1017.61	32461.759	5.6480	3.1715
32.0	100.53	804.25	1024.00	32768.000	5.6569	3.1748
32.1	100.85	809.28	1030.41	33076.161	5.6656	3.1781
32.2	101.16	814.33	1036.84	33386.248	5.6745	3.1814
32.3	101.47	819.40	1043.29	33698.267	5.6833	3.1847
32.4	101.79	824.48	1049.76	34012.224	5.6921	3.1880
32.5	102.10	829.58	1056.25	34328.125	5.7008	3.1913
32.6	102.42	834.69	1062.76	34645.976	5.7096	3.1945
32.7	102.73	839.82	1069.29	34965.783	5.7183	3.1978
32.8	103.04	844.96	1075.84	35287.552	5.7271	3.2010
32.9	103.36	850.12	1082.41	35611.289	5.7358	3.2043
33.0	103.67	855.30	1089.00	35937.000	5.7446	3.2075
33.1	103.99	860.49	1095.61	36264.691	5.7532	3.2108
33.2	104.30	865.70	1102.24	36594.368	5.7619	3.2140
33.3	104.62	870.92	1108.89	36926.037	5.7706	3.2172
33.4	104.93	876.16	1115.56	37259.704	5.7792	3.2204
33.5	105.24	881.41	1122.25	37595.375	5.7879	3.2237
33.6	105.56	886.68	1128.96	37933.056	5.7965	3.2269
33.7	105.87	891.97	1135.69	38272.753	5.8051	3.2301
33.8	106.19	897.27	1142.44	38614.472	5.8137	3.2332
33.9	106.50	902.59	1149.21	38958.219	5.8223	3.2364
34.0	106.81	907.92	1156.00	39304.000	5.8310	3.2396
34.1	107.13	913.27	1162.81	39651.821	5.8395	3.2428
34.2	107.44	918.63	1169.64	40001.688	5.8480	3.2460
34.3	107.76	924.01	1176.49	40353.607	5.8566	3.2491
34.4	108.07	929.41	1183.36	40707.584	5.8651	3.2522

CONSTANTS—*Continued.*

n	$n\pi$	$n^2\frac{\pi}{4}$	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$
34.5	108.38	934.82	1190.25	41063.625	5.8736	3.2554
34.6	108.70	940.25	1197.16	41421.736	5.8821	3.2586
34.7	109.01	945.69	1204.09	41781.923	5.8906	3.2617
34.8	109.33	951.15	1211.04	42144.192	5.8991	3.2648
34.9	109.64	956.62	1218.01	42508.549	5.9076	3.2679
35.0	109.96	962.11	1225.00	42875.000	5.9161	3.2710
35.1	110.27	967.62	1232.01	43243.551	5.9245	3.2742
35.2	110.58	973.14	1239.04	43614.208	5.9329	3.2773
35.3	110.90	978.68	1246.09	43986.977	5.9413	3.2804
35.4	111.21	984.23	1253.16	44361.864	5.9497	3.2835
35.5	111.53	989.80	1260.25	44738.875	5.9581	3.2866
35.6	111.84	995.38	1267.36	45118.016	5.9665	3.2897
35.7	112.15	1000.98	1274.49	45499.293	5.9749	3.2927
35.8	112.47	1006.60	1281.64	45882.712	5.9833	3.2958
35.9	112.78	1012.23	1288.81	46268.279	5.9916	3.2989
36.0	113.10	1017.88	1296.00	46656.000	6.0000	3.3019
36.1	113.41	1023.54	1303.21	47045.881	6.0083	3.3050
36.2	113.73	1029.22	1310.44	47437.928	6.0166	3.3080
36.3	114.04	1034.91	1317.69	47832.147	6.0249	3.3111
36.4	114.35	1040.62	1324.96	48228.544	6.0332	3.3141
39.5	114.67	1046.35	1332.25	48627.125	6.0415	3.3171
36.6	114.98	1052.09	1339.56	49027.896	6.0497	3.3202
36.7	115.30	1057.84	1346.89	49430.863	6.0580	3.3232
36.8	115.61	1063.62	1354.24	49836.032	6.0663	3.3262
36.9	115.92	1069.41	1361.61	50243.409	6.0745	3.3292
37.0	116.24	1075.21	1369.00	50653.000	6.0827	3.3322
37.1	116.55	1081.03	1376.41	51064.811	6.0909	3.3352
37.2	116.87	1086.87	1383.84	51478.848	6.0991	3.3382
37.3	117.18	1092.72	1391.29	51895.117	6.1073	3.3412
37.4	117.50	1098.58	1398.76	52313.624	6.1155	3.3442
37.5	117.81	1104.47	1406.25	52734.375	6.1237	3.3472
37.6	118.12	1110.36	1413.76	53157.376	6.1318	3.3501
37.7	118.44	1116.28	1421.29	53582.633	6.1400	3.3531
37.8	118.75	1122.21	1428.84	54010.152	6.1481	3.3561
37.9	119.07	1128.15	1436.41	54439.939	6.1563	3.3590
38.0	119.38	1134.11	1444.00	54872.000	6.1644	3.3620
38.1	119.69	1140.09	1451.61	55306.341	6.1725	3.3649
38.2	120.01	1146.08	1459.24	55742.968	6.1806	3.3679
38.3	120.32	1152.09	1466.89	56181.887	6.1887	3.3708
38.4	120.64	1158.12	1474.56	56623.104	6.1967	3.3737
38.5	120.95	1164.16	1482.25	57066.625	6.2048	3.3767
38.6	121.27	1170.21	1489.96	57512.456	6.2129	3.3796
38.7	121.58	1176.28	1497.69	57960.603	6.2209	3.3825

CONSTANTS—*Continued.*

n	πn	$n^2\frac{\pi}{4}$	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$
38.8	121.89	1182.37	1505.44	58411.072	6.2289	3.3854
38.9	122.21	1188.47	1513.21	58863.869	6.2370	3.3883
39.0	122.52	1194.59	1521.00	59319.000	6.2450	3.3912
39.1	122.84	1200.72	1528.81	59776.471	6.2530	3.3941
39.2	123.15	1206.87	1536.64	60236.288	6.2610	3.3970
39.3	123.46	1213.04	1544.49	60698.457	6.2689	3.3999
39.4	123.78	1219.22	1552.36	61162.984	6.2769	3.4028
39.5	124.09	1225.42	1560.25	61629.875	6.2849	3.4056
39.6	124.41	1231.63	1568.16	62099.136	6.2928	3.4085
39.7	124.72	1237.86	1576.09	62570.773	6.3008	3.4114
39.8	125.04	1244.10	1584.04	63044.792	6.3087	3.4142
39.9	125.35	1250.36	1592.01	63521.199	6.3166	3.4171
40.0	125.66	1256.64	1600.00	64000.000	6.3245	3.4200
40.1	125.98	1262.93	1608.01	64481.201	6.3325	3.4228
40.2	126.29	1269.23	1616.04	64964.808	6.3404	3.4256
40.3	126.61	1275.56	1624.09	65450.827	6.3482	3.4285
40.4	126.92	1281.90	1632.16	65939.264	6.3561	3.4313
40.5	127.23	1288.25	1640.25	66430.125	6.3639	3.4341
40.6	127.55	1294.62	1648.36	66923.416	6.3718	3.4370
40.7	127.86	1301.00	1656.49	67419.143	6.3796	3.4398
40.8	128.18	1307.41	1664.64	67911.312	6.3875	3.4426
40.9	128.49	1313.82	1672.81	68417.929	6.3953	3.4454
41.0	128.81	1320.25	1681.00	68921.000	6.4031	3.4482
41.1	129.12	1326.70	1689.21	69426.531	6.4109	3.4510
41.2	129.43	1333.17	1697.44	69934.528	6.4187	3.4538
41.3	129.75	1339.65	1705.69	70444.997	6.4265	3.4566
41.4	130.06	1346.14	1713.96	70957.944	6.4343	3.4594
41.5	130.38	1352.65	1722.25	71473.375	6.4421	3.4622
41.6	130.69	1359.18	1730.56	71991.296	6.4498	3.4650
41.7	131.00	1365.72	1738.89	72511.713	6.4575	3.4677
41.8	131.32	1372.28	1747.24	73034.632	6.4653	3.4705
41.9	131.63	1378.85	1755.61	73560.059	6.4730	3.4733
42.0	131.95	1385.44	1764.00	74088.000	6.4807	3.4760
42.1	132.26	1392.05	1772.41	74618.461	6.4884	3.4788
42.2	132.58	1398.67	1780.84	75151.448	6.4961	3.4815
42.3	132.89	1405.31	1789.29	75686.967	6.5038	3.4843
42.4	133.20	1411.96	1797.76	76225.024	6.5115	3.4870
42.5	133.52	1418.63	1806.25	76765.625	6.5192	3.4898
42.6	133.83	1425.31	1814.76	77308.776	6.5268	3.4925
42.7	134.15	1432.01	1823.29	77854.483	6.5345	3.4952
42.8	134.46	1438.72	1831.84	78402.752	6.5422	3.4080
42.9	134.77	1445.45	1840.41	78953.589	6.5498	3.5007

CONSTANTS—*Continued.*

n	$n\pi$	$n^2\frac{\pi}{4}$	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$
43.0	135.09	1452.20	1849.00	79507.000	6.5574	3.5034
43.1	135.40	1458.96	1857.61	80062.991	6.5651	3.5061
43.2	135.72	1465.74	1866.24	80621.568	6.5727	3.5088
43.3	136.03	1472.54	1874.89	81182.737	6.5803	3.5115
43.4	136.35	1479.34	1883.56	81746.504	6.5879	3.5142
43.5	136.66	1486.17	1892.25	82312.875	6.5954	3.5169
43.6	136.97	1493.01	1900.96	82881.856	6.6030	3.5196
43.7	137.29	1499.87	1909.69	83453.453	6.6106	3.5223
43.8	137.60	1506.74	1918.44	84027.672	6.6182	3.5250
43.9	137.92	1513.63	1927.21	84604.519	6.6257	3.5277
44.0	138.23	1520.53	1936.00	85184.000	6.6333	3.5303
44.1	138.54	1527.45	1944.81	85766.121	6.6408	3.5330
44.2	138.86	1534.39	1953.64	86350.888	6.6483	3.5357
44.3	139.17	1541.34	1962.49	86938.307	6.6558	3.5384
44.4	139.49	1548.30	1971.36	87528.384	6.6633	3.5410
44.5	139.80	1555.28	1980.25	88121.125	6.6708	3.5437
44.6	140.12	1562.28	1989.16	88716.536	6.6783	3.5463
44.7	140.43	1569.30	1998.09	89314.623	6.6858	3.5490
44.8	140.74	1576.33	2007.04	89915.392	6.6933	3.5516
44.9	141.06	1583.37	2016.01	90518.849	6.7007	3.5543
45.0	141.37	1590.43	2025.00	91125.000	6.7082	3.5569
45.1	141.69	1597.51	2034.01	91733.851	6.7156	3.5595
45.2	142.00	1604.60	2043.04	92345.408	6.7231	3.5621
45.3	142.31	1611.71	2052.09	92959.677	6.7305	3.5648
45.4	142.63	1618.83	2061.16	93576.664	6.7379	3.5674
45.5	142.94	1625.97	2070.25	94196.375	6.7454	3.5700
45.6	143.26	1633.13	2079.36	94818.816	6.7528	3.5726
45.7	143.57	1640.30	2088.49	95443.993	6.7602	3.5752
45.8	143.88	1647.48	2097.64	96071.912	6.7676	3.5778
45.9	144.20	1654.68	2106.81	96702.579	6.7749	3.5805
46.0	144.51	1661.90	2116.00	97336.000	6.7823	3.5830
46.1	144.83	1669.14	2125.21	97972.181	6.7897	3.5856
46.2	145.14	1676.39	2134.44	98611.128	6.7971	3.5882
46.3	145.46	1683.65	2143.69	99252.847	6.8044	3.5908
46.4	145.77	1690.93	2152.96	99897.344	6.8117	3.5934
46.5	146.08	1698.23	2162.25	100544.625	6.8191	3.5960
46.6	146.40	1705.54	2171.56	101194.696	6.8264	3.5986
46.7	146.71	1712.87	2180.89	101847.563	6.8337	3.6011
46.8	147.03	1720.21	2190.24	102503.232	6.8410	3.6037
46.9	147.34	1727.57	2199.61	103161.709	6.8484	3.6063
47.0	147.65	1734.94	2209.00	103823.000	6.8556	3.6088
47.1	147.97	1742.34	2218.41	104487.111	6.8629	3.6114
47.2	148.28	1749.74	2227.84	105154.048	6.8702	3.6139

CONSTANTS—*Continued.*

n	πn	$n^2\frac{\pi}{4}$	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$
47.3	148.60	1757.16	2237.29	105823.817	6.8775	3.6164
47.4	148.91	1764.60	2246.76	106496.424	6.8848	3.6190
47.5	149.23	1772.05	2256.25	107171.875	6.8920	3.6216
47.6	149.54	1779.52	2265.76	107850.176	6.8993	3.6241
47.7	149.85	1787.01	2275.29	108531.333	6.9065	3.6267
47.8	150.17	1794.51	2284.84	109215.352	6.9137	3.6292
47.9	150.48	1802.03	2294.41	109902.239	6.9209	3.6317
48.0	150.80	1809.56	2304.00	110592.000	6.9282	3.6342
48.1	151.11	1817.11	2313.61	111284.641	6.9354	3.6368
48.2	151.42	1824.67	2323.24	111980.168	6.9426	3.6393
48.3	151.74	1832.25	2332.89	112678.587	6.9498	3.6418
48.4	152.05	1839.84	2342.56	113379.904	6.9570	3.6443
48.5	152.37	1847.45	2352.25	114084.125	6.9642	3.6468
48.6	152.68	1855.08	2361.96	114791.256	6.9714	3.6493
48.7	153.00	1862.72	2371.69	115501.303	6.9785	3.6518
48.8	153.31	1870.38	2381.44	116214.272	6.9857	3.6543
48.9	153.62	1878.05	2391.21	116930.169	6.9928	3.6568
49.0	153.94	1885.74	2401.00	117649.000	7.0000	3.6593
49.1	154.25	1893.45	2410.81	118370.771	7.0071	3.6618
49.2	154.57	1901.17	2420.64	119095.488	7.0143	3.6643
49.3	154.88	1908.90	2430.49	119823.157	7.0214	3.6668
49.4	155.19	1916.65	2440.36	120553.784	7.0285	3.6692
49.5	155.51	1924.42	2450.25	121287.375	7.0356	3.6717
49.6	155.82	1932.21	2460.16	122023.936	7.0427	3.6742
49.7	156.14	1940.00	2470.09	122763.473	7.0498	3.6767
49.8	156.45	1947.82	2480.04	123505.992	7.0569	3.6791
49.9	156.77	1955.65	2490.01	124251.499	7.0640	3.6816
50.0	157.08	1963.50	2500.00	125000.000	7.0711	3.6840
51.0	160.22	2042.82	2601.00	132651.000	7.1414	3.7084
52.0	163.36	2123.72	2704.00	140608.000	7.2111	3.7325
53.0	166.50	2206.19	2809.00	148877.000	7.2801	3.7563
54.0	169.64	2290.22	2916.00	157464.000	7.3485	3.7798
55.0	172.78	2375.83	3025.00	166375.000	7.4162	3.8030
56.0	175.93	2463.01	3136.00	175616.000	7.4833	3.8259
57.0	179.07	2551.76	3249.00	185193.000	7.5498	3.8485
58.0	182.21	2642.08	3364.00	195112.000	7.6158	3.8709
59.0	185.35	2733.97	3481.00	205379.000	7.6811	3.8930
60.0	188.49	2827.44	3600.00	216000.000	7.7460	3.9149
61.0	191.63	2922.47	3721.00	226981.000	7.8102	3.9365
62.0	194.77	3019.07	3844.00	238328.000	7.8740	3.9579
63.0	197.92	3117.25	3969.00	250047.000	7.9373	3.9791
64.0	201.06	3216.99	4096.00	262144.000	8.0000	4.0000
65.0	204.20	3318.31	4225.00	274625.000	8.0623	4.0207
66.0	207.34	3421.20	4356.00	287496.000	8.1240	4.0412

CONSTANTS—*Continued.*

n	$n\pi$	$n^2\frac{\pi}{4}$	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$
67.0	210.48	3525.66	4489.00	300763.000	8.1854	4.0615
68.0	213.63	3631.69	4624.00	314432.000	8.2462	4.0817
69.0	216.77	3739.29	4761.00	328509.000	8.3066	4.1016
70.0	219.91	3848.46	4900.00	343000.000	8.3666	4.1213
71.0	223.05	3959.20	5041.00	357911.000	8.4261	4.1408
72.0	226.19	4071.51	5184.00	373248.000	8.4853	4.1602
73.0	229.33	4185.39	5329.00	389017.000	8.5440	4.1793
74.0	232.47	4300.85	5476.00	405224.000	8.6023	4.1983
75.0	235.62	4417.87	5625.00	421875.000	8.6603	4.2172
76.0	238.76	4536.47	5776.00	438976.000	8.7178	4.2358
77.0	241.90	4656.63	5929 00	456533.000	8.7750	4.2543
78.0	245.04	4778.37	6084.00	474552.000	8.8318	4.2727
79.0	248.18	4901.68	6241.00	493039.000	8.8882	4.2908
80.0	251.32	5026.56	6400.00	512000.000	8.9443	4.3089
81.0	254.47	5153.01	6561.00	531441.000	9.0000	4.3267
82.0	257.61	5281.03	6724.00	551368.000	9.0554	4.3445
83.0	260.75	5410.62	6889.00	571787.000	9.1104	4.3621
84.0	263.89	5541.78	7056.00	592704.000	9.1652	4.3795
85.0	267.03	5674.50	7225.00	614125.000	9.2195	4.3968
86.0	270.17	5808.81	7396.00	636056.000	9.2736	4.4140
87.0	273.32	5944.69	7569.00	658503.000	9.3274	4.4310
88.0	276.46	6082.13	7744.00	681472.000	9.3808	4.4480
89.0	279.60	6221.13	7921.00	704969.000	9.4340	4.4647
90.0	282.74	6361.74	8100.00	729000.000	9.4868	4.4814
91.0	285.88	6503.89	8281.00	753571.000	9.5394	4.4979
92.0	289.02	6647.62	8464.00	778688.000	9.5917	4.5144
93.0	292.17	6792.92	8649.00	804357.000	9.6437	4.5307
94.0	295.31	6939.78	8836.00	830584.000	9.6954	4.5468
95.0	298.45	7088.23	9025.00	857375.000	9.7468	4.5629
96.0	301.59	7238.24	9216.00	884736.000	9.7980	4.5789
97.0	304.73	7389.83	9409.00	912673.000	9.8489	4.5947
98.0	307.87	7542.98	9604.00	941192.000	9.8995	4.6104
99.0	311.02	7697.68	9801.00	970299.000	9.9499	4.6261
100.0	314.16	7854.00	10000.00	1000000.000	10.0000	4.6416

II.

LOGARITHMS.

Hyperbolic Logarithms.

N.	Log.	N.	Log.	N.	Log.	N.	Log.	N.	Log.
1.00	0.0000	2.30	0.8329	3.60	1.2809	4.90	1.5892	6.40	1.8563
1.05	0.0488	2.35	0.8544	3.65	1.2947	4.95	1.5994	6.50	1.8718
1.10	0.0953	2.40	0.8755	3.70	1.3083	5.00	1.6094	6.60	1.8871
1.15	0.1398	2.45	0.8961	3.75	1.3218	5.05	1.6194	6.70	1.9021
1.20	0.1823	2.50	0.9163	3.80	1.3350	5.10	1.6292	6.80	1.9169
1.25	0.2231	2.55	0.9361	3.85	1.3481	5.15	1.6390	6.90	1.9315
1.30	0.2624	2.60	0.9555	3.90	1.3610	5.20	1.6487	7.00	1.9459
1.35	0.3001	2.65	0.9746	3.95	1.3737	5.25	1.6582	7.20	1.9741
1.40	0.3365	2.70	0.9933	4.00	1.3863	5.30	1.6677	7.40	2.0015
1.45	0.3716	2.75	1.0116	4.05	1.3987	5.35	1.6771	7.60	2.0281
1.50	0.4055	2.80	1.0296	4.10	1.4110	5.40	1.6864	7.80	2.0541
1.55	0.4383	2.85	1.0473	4.15	1.4231	5.45	1.6956	8.00	2.0794
1.60	0.4700	2.90	1.0647	4.20	1.4351	5.50	1.7047	8.25	2.1102
1.65	0.5008	2.95	1.0818	4.25	1.4469	5.55	1.7138	8.50	2.1401
1.70	0.5306	3.00	1.0986	4.30	1.4586	5.60	1.7228	8.75	2.1691
1.75	0.5596	3.05	1.1154	4.35	1.4701	5.65	1.7317	9.00	2.1972
1.80	0.5878	3.10	1.1314	4.40	1.4816	5.70	1.7405	9.25	2.2246
1.85	0.6152	3.15	1.1474	4.45	1.4929	5.75	1.7492	9.50	2.2513
1.90	0.6419	3.20	1.1632	4.50	1.5041	5.80	1.7579	9.75	2.2773
1.95	0.6678	3.25	1.1787	4.55	1.5151	5.85	1.7664	10.00	2.3026
2.00	0.6931	3.30	1.1939	4.60	1.5261	5.90	1.7750	11.00	2.3979
2.05	0.7178	3.35	1.2090	4.65	1.5369	5.95	1.7834	12.00	2.4849
2.10	0.7419	3.40	1.2238	4.70	1.5476	6.00	1.7918	13.00	2.5649
2.15	0.7655	3.45	1.2384	4.75	1.5581	6.10	1.8083	14.00	2.6391
2.20	0.7885	3.50	1.2528	4.80	1.5686	6.20	1.8245	15.00	2.7081
2.25	0.8109	3.55	1.2669	4.85	1.5790	6.30	1.8405	16.00	2.7726

Common Logarithms: 10–1200.

N.	0	1	2	3	4	5	6	7	8	9	Diff.
10	00000	00432	00860	01284	01703	02119	02531	02938	03342	03743	396
11	04139	04532	04922	05308	05690	06070	06446	06819	07188	07555	363
12	07918	08279	08636	08991	09342	09691	10037	10380	10721	11059	335
13	11394	11727	12057	12385	12710	13033	13354	13672	13988	14301	312
14	14613	14922	15229	15534	15836	16137	16435	16732	17026	17319	290
15	17609	17898	18184	18469	18752	19033	19312	19590	19866	20140	272
16	20412	20683	20952	21219	21484	21748	22011	22272	22531	22789	256
17	23045	23300	23553	23805	24055	24304	24551	24797	25042	25285	242
18	25527	25768	26007	26245	26482	26717	26951	27184	27416	27646	229
19	27875	28103	28330	28556	28780	29003	29226	29447	29667	29885	218
20	30103	30320	30535	30750	30963	31175	31387	31597	31806	32015	207
21	32222	32428	32634	32838	33041	33244	33445	33646	33846	34044	198
22	34242	34439	34635	34830	35025	35218	35411	35603	35793	35984	189
23	36173	36361	36549	36736	36922	37107	37291	37475	37658	37840	181
24	38021	38202	38382	38561	38739	38917	39094	39270	39445	39620	174
25	39794	39967	40140	40312	40483	40654	40824	40993	41162	41330	167
26	41497	41664	41830	41996	42160	42325	42488	42651	42813	42975	161
27	43136	43297	43457	43616	43775	43933	44091	44248	44404	44560	156
28	44716	44871	45025	45179	45332	45484	45637	45788	45939	46090	150
29	46240	46389	46538	46687	46835	46982	47129	47276	47422	47567	145

CONSTANTS—*Continued.*

n	$n\pi$	$n^2\frac{\pi}{4}$	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$
67.0	210.48	3525.66	4489.00	300763.000	8.1854	4.0615
68.0	213.63	3631.69	4624.00	314432.000	8.2462	4.0817
69.0	216.77	3739.29	4761.00	328509.000	8.3066	4.1016
70.0	219.91	3848.46	4900.00	343000.000	8.3666	4.1213
71.0	223.05	3959.20	5041.00	357911.000	8.4261	4.1408
72.0	226.19	4071.51	5184.00	373248.000	8.4853	4.1602
73.0	229.33	4185.39	5329.00	389017.000	8.5440	4.1793
74.0	232.47	4300.85	5476.00	405224.000	8.6023	4.1983
75.0	235.62	4417.87	5625.00	421875.000	8.6603	4.2172
76.0	238.76	4536.47	5776.00	438976.000	8.7178	4.2358
77.0	241.90	4656.63	5929 00	456533.000	8.7750	4.2543
78.0	245.04	4778.37	6084.00	474552.000	8.8318	4.2727
79.0	248.18	4901.68	6241.00	493039.000	8.8882	4.2908
80.0	251.32	5026.56	6400.00	512000.000	8.9443	4.3089
81.0	254.47	5153.01	6561.00	531441.000	9.0000	4.3267
82.0	257.61	5281.03	6724.00	551368.000	9.0554	4.3445
83.0	260.75	5410.62	6889.00	571787.000	9.1104	4.3621
84.0	263.89	5541.78	7056.00	592704.000	9.1652	4.3795
85.0	267.03	5674.50	7225.00	614125.000	9.2195	4.3968
86.0	270.17	5808.81	7396.00	636056.000	9.2736	4.4140
87.0	273.32	5944.69	7569.00	658503.000	9.3274	4.4310
88.0	276.46	6082.13	7744.00	681472.000	9.3808	4.4480
89.0	279.60	6221.13	7921.00	704969.000	9.4340	4.4647
90.0	282.74	6361.74	8100.00	729000.000	9.4868	4.4814
91.0	285.88	6503.89	8281.00	753571.000	9.5394	4.4979
92.0	289.02	6647.62	8464.00	778688.000	9.5917	4.5144
93.0	292.17	6792.92	8649.00	804357.000	9.6437	4.5307
94.0	295.31	6939.78	8836.00	830584.000	9.6954	4.5468
95.0	298.45	7088.23	9025.00	857375.000	9.7468	4.5629
96.0	301.59	7238.24	9216.00	884736.000	9.7980	4.5789
97.0	304.73	7389.83	9409.00	912673.000	9.8489	4.5947
98.0	307.87	7542.98	9604.00	941192.000	9.8995	4.6104
99.0	311.02	7697.68	9801.00	970299.000	9.9499	4.6261
100.0	314.16	7854.00	10000.00	1000000.000	10.0000	4.6416

II.

LOGARITHMS.

Hyperbolic Logarithms.

N.	Log.	N.	Log.	N.	Log.	N.	Log.	N.	Log.
1.00	0.0000	2.30	0.8329	3.60	1.2809	4.90	1.5892	6.40	1.8563
1.05	0.0488	2.35	0.8544	3.65	1.2947	4.95	1.5994	6.50	1.8718
1.10	0.0953	2.40	0.8755	3.70	1.3083	5.00	1.6094	6.60	1.8871
1.15	0.1398	2.45	0.8961	3.75	1.3218	5.05	1.6194	6.70	1.9021
1.20	0.1823	2.50	0.9163	3.80	1.3350	5.10	1.6292	6.80	1.9169
1.25	0.2231	2.55	0.9361	3.85	1.3481	5.15	1.6300	6.90	1.9315
1.30	0.2624	2.60	0.9555	3.90	1.3610	5.20	1.6487	7.00	1.9459
1.35	0.3001	2.65	0.9746	3.95	1.3737	5.25	1.6582	7.20	1.9741
1.40	0.3365	2.70	0.9933	4.00	1.3863	5.30	1.6677	7.40	2.0015
1.45	0.3716	2.75	1.0116	4.05	1.3987	5.35	1.6771	7.60	2.0281
1.50	0.4055	2.80	1.0296	4.10	1.4110	5.40	1.6864	7.80	2.0541
1.55	0.4383	2.85	1.0473	4.15	1.4231	5.45	1.6956	8.00	2.0794
1.60	0.4700	2.90	1.0647	4.20	1.4351	5.50	1.7047	8.25	2.1102
1.65	0.5008	2.95	1.0818	4.25	1.4469	5.55	1.7138	8.50	2.1401
1.70	0.5306	3.00	1.0986	4.30	1.4586	5.60	1.7228	8.75	2.1691
1.75	0.5596	3.05	1.1154	4.35	1.4701	5.65	1.7317	9.00	2.1972
1.80	0.5878	3.10	1.1314	4.40	1.4816	5.70	1.7405	9.25	2.2246
1.85	0.6152	3.15	1.1474	4.45	1.4929	5.75	1.7402	9.50	2.2513
1.90	0.6419	3.20	1.1632	4.50	1.5041	5.80	1.7579	9.75	2.2773
1.95	0.6678	3.25	1.1787	4.55	1.5151	5.85	1.7664	10.00	2.3026
2.00	0.6931	3.30	1.1939	4.60	1.5261	5.90	1.7750	11.00	2.3979
2.05	0.7178	3.35	1.2090	4.65	1.5369	5.95	1.7834	12.00	2.4849
2.10	0.7419	3.40	1.2238	4.70	1.5476	6.00	1.7918	13.00	2.5649
2.15	0.7655	3.45	1.2384	4.75	1.5581	6.10	1.8083	14.00	2.6391
2.20	0.7885	3.50	1.2528	4.80	1.5686	6.20	1.8245	15.00	2.7081
2.25	0.8109	3.55	1.2669	4.85	1.5790	6.30	1.8405	16.00	2.7726

Common Logarithms: 10–1200.

N.	0	1	2	3	4	5	6	7	8	9	Diff.
10	00000	00432	00860	01284	01703	02119	02531	02938	03342	03743	396
11	04139	04532	04922	05308	05690	06070	06446	06819	07188	07555	363
12	07918	08279	08636	08991	09342	09691	10037	10380	10721	11059	335
13	11394	11727	12057	12385	12710	13033	13354	13672	13988	14301	312
14	14613	14922	15229	15534	15836	16137	16435	16732	17026	17319	290
15	17609	17898	18184	18469	18752	19033	19312	19590	19866	20140	272
16	20412	20683	20952	21219	21484	21748	22011	22272	22531	22789	256
17	23045	23300	23553	23805	24055	24304	24551	24797	25042	25285	242
18	25527	25768	26007	26245	26482	26717	26951	27184	27416	27646	229
19	27875	28103	28330	28556	28780	29003	29226	29447	29667	29885	218
20	30103	30320	30535	30750	30963	31175	31387	31597	31806	32015	207
21	32222	32428	32634	32838	33041	33244	33445	33646	33846	34044	198
22	34242	34439	34635	34830	35025	35218	35411	35603	35793	35984	189
23	36173	36361	36549	36736	36922	37107	37291	37475	37658	37840	181
24	38021	38202	38382	38561	38739	38917	39094	39270	39445	39620	174
25	39794	39967	40140	40312	40483	40654	40824	40993	41162	41330	167
26	41497	41664	41830	41906	42160	42325	42488	42651	42813	42975	161
27	43136	43297	43457	43616	43775	43933	44091	44248	44404	44560	156
28	44716	44871	45025	45179	45332	45484	45637	45788	45930	46090	150
29	46240	46389	46538	46687	46835	46982	47129	47276	47422	47567	145

COMMON LOGARITHMS—*Continued.*

N.	0	1	2	3	4	5	6	7	8	9	Dif.
30	47712	47857	48001	48144	48287	48430	48572	48714	48855	48996	140
31	49136	49276	49415	49554	49693	49831	49969	50106	50243	50379	136
32	50515	50651	50786	50920	51055	51188	51322	51455	51587	51720	130
33	51851	51983	52114	52244	52375	52504	52634	52763	52892	53020	128
34	53148	53275	53403	53529	53656	53782	53908	54033	54158	54283	124
35	54407	54531	54654	54777	54900	55023	55145	55267	55388	55509	121
36	55630	55751	55871	55991	56110	56229	56348	56467	56585	56703	117
37	56820	56937	57054	57171	57287	57403	57519	57634	57749	57864	114
38	57978	58092	58206	58320	58433	58546	58659	58771	58883	58995	111
39	59106	59218	59329	59439	59550	59660	59770	59879	59988	60097	109
40	60206	60314	60423	60531	60638	60746	60853	60959	61066	61172	106
41	61278	61384	61490	61595	61700	61805	61909	62014	62118	62221	104
42	62325	62428	62531	62634	62737	62839	62941	63043	63144	63246	102
43	63347	63448	63548	63649	63749	63849	63949	64048	64147	64246	99
44	64345	64444	64542	64640	64738	64836	64933	65031	65128	65225	97
45	65321	65418	65514	65610	65706	65801	65896	65992	66087	66181	95
46	66276	66370	66464	66558	66652	66745	66839	66932	67025	67117	93
47	67210	67302	67394	67486	67578	67669	67761	67852	67943	68034	91
48	68124	68215	68305	68395	68485	68574	68664	68753	68842	68931	89
49	69020	69108	69197	69285	69373	69461	69548	69636	69723	69810	87
50	69897	69984	70070	70157	70243	70329	70415	70501	70586	70672	86
51	70757	70842	70927	71012	71096	71181	71265	71349	71433	71517	84
52	71600	71684	71767	71850	71933	72016	72099	72181	72263	72346	83
53	72428	72509	72591	72673	72754	72835	72916	72997	73078	73159	81
54	73239	73320	73400	73480	73560	73640	73719	73799	73878	73957	80
55	74036	74115	74194	74273	74351	74429	74507	74586	74663	74741	78
56	74819	74896	74974	75051	75128	75205	75282	75358	75435	75511	77
57	75587	75664	75740	75815	75891	75967	76042	76118	76193	76268	76
58	76343	76418	76492	76567	76641	76716	76790	76864	76938	77012	74
59	77085	77159	77232	77305	77379	77452	77525	77597	77670	77743	73
60	77815	77887	77960	78032	78104	78176	78247	78319	78390	78462	72
61	78533	78604	78675	78746	78817	78888	78958	79029	79099	79169	71
62	79239	79309	79379	79449	79518	79588	79657	79727	79796	79865	70
63	79934	80003	80072	80140	80209	80277	80346	80414	80482	80550	68
64	80618	80686	80754	80821	80889	80956	81023	81090	81158	81224	67
65	81291	81358	81425	81491	81558	81624	81690	81757	81823	81889	66
66	81954	82020	82086	82151	82217	82282	82347	82413	82478	82543	65
67	82607	82672	82737	82802	82866	82930	82995	83059	83123	83187	64
68	83251	83315	83378	83442	83506	83569	83632	83696	83759	83822	63
69	83885	83948	84011	84073	84136	84198	84261	84323	84386	84448	63
70	84510	84572	84634	84696	84757	84819	84880	84942	85003	85065	62
71	85126	85187	85248	85309	85370	85431	85491	85552	85612	85673	61
72	85733	85794	85854	85914	85974	86034	86094	86153	86213	86273	60
73	86332	86392	86451	86510	86570	86629	86688	86747	86806	86864	59
74	86923	86982	87040	87099	87157	87216	87274	87332	87390	87448	58
75	87506	87564	87622	87679	87737	87795	87852	87910	87967	88024	58
76	88081	88138	88195	88252	88309	88366	88423	88480	88536	88593	57
77	88649	88705	88762	88818	88874	88930	88986	89042	89098	89154	56
78	89209	89265	89321	89376	89432	89487	89542	89597	89653	89708	55
79	89763	89818	89873	89927	89982	90037	90091	90146	90200	90255	55
80	90309	90363	90417	90472	90526	90580	90634	90687	90741	90795	54
81	90849	90902	90956	91009	91062	91116	91169	91222	91275	91328	53
82	91381	91434	91487	91540	91593	91645	91698	91751	91803	91855	52
83	91908	91960	92012	92065	92117	92169	92221	92273	92324	92376	52
84	92428	92480	92531	92583	92634	92686	92737	92788	92840	92891	51
85	92942	92993	93044	93095	93146	93197	93247	93298	93349	93399	51
86	93450	93500	93551	93601	93651	93702	93752	93802	93852	93902	50
87	93952	94002	94052	94101	94151	94201	94250	94300	94349	94399	50

Common Logarithms—Continued

	1	2	3	4	5	6	7	8	9	Diff
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	97359	[illegible]	[illegible]	[illegible]	97543	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	99607	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	00043	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	43
	00475	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	43
	00903	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	42
	01328	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	42
	01745	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	42
	02160	[illegible]	02243	02284	02325	[illegible]	[illegible]	[illegible]	[illegible]	41
	02572	[illegible]	[illegible]	02694	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	41
	02979	[illegible]	[illegible]	03100	03141	03181	03222	[illegible]	[illegible]	41
	03383	03423	[illegible]	03503	03543	03583	03623	[illegible]	[illegible]	40
	03782	03822	[illegible]	[illegible]	03941	03981	04021	[illegible]	[illegible]	40
	04179	04218	04258	04297	04336	[illegible]	04415	04454	04493	39
	04571	[illegible]	[illegible]	04689	04727	[illegible]	04805	04844	04883	39
	04961	[illegible]	05038	05077	05115	05154	05192	05231	[illegible]	39
	05346	05385	05423	05461	05500	05538	[illegible]	05614	05652	38
	05729	05767	05805	05843	05881	05918	05956	[illegible]	[illegible]	38
0	06108	06145	06183	06221	06258	06296	06333	06371	[illegible]	38
6	06483	06521	06558	06595	06633	06670	[illegible]	06744	06781	37
9	06856	06893	06930	06967	07004	07041	07078	07115	07151	37
8	07225	07262	07298	07335	07372	07408	07445	07482	07518	37
5	07591	07628	07664	07700	07773	07773	07809	07846	07882	36

		Log.
Naperian logarithms, ...	$e = 2.7182818$	0.4342945
Modulus of common logarithms, ...	$M = 0.4342945$	9.6377843 − 10

III.

MEAN PRESSURE RATIOS. (NORTHCOTT.)

r	A	B	C	r	A	B	C	r	A	B	C	r	A	B	C
1.0	1.000	1.000	1.000	5.3	.478	.503	.488	9.6	.312	.340	.324	17.8	.194	.218	.204
1.1	0.996	0.996	0.996	5.4	.472	.497	.482	9.7	.310	.338	.322	18.0	.192	.216	.202
1.2	0.983	0.983	0.983	5.5	.467	.492	.477	9.8	.307	.335	.319	18.2	.190	.215	.200
1.3	.966	.968	.967	5.6	.461	.486	.471	9.9	.305	.333	.317	18.4	.189	.214	.199
1.4	.947	.952	.950	5.7	.456	.481	.466	10.0	.303	.330	.314	18.6	.187	.212	.197
1.5	.928	.934	.931	5.8	.450	.475	.460	10.2	.299	.325	.310	18.8	.185	.210	.195
1.6	.910	.919	.914	5.9	.445	.470	.455	10.4	.295	.321	.306	19.0	.183	.208	.193
1.7	.890	.900	.895	6.0	.440	.465	.450	10.6	.291	.317	.302	19.2	.182	.207	.192
1.8	.870	.880	.875	6.1	.434	.460	.445	10.8	.287	.313	.298	19.4	.180	.205	.190
1.9	.850	.862	.856	6.2	.429	.455	.440	11.0	.283	.309	.294	19.6	.179	.204	.189
2.0	.833	.846	.840	6.3	.424	.450	.435	11.2	.279	.305	.290	19.8	.178	.202	.187
2.1	.817	.830	.824	6.4	.419	.445	.430	11.4	.275	.301	.286	20.0	.177	.200	.186
2.2	.798	.812	.805	6.5	.414	.441	.426	11.6	.272	.298	.283	20.2	.175	.198	.184
2.3	.780	.795	.787	6.6	.409	.436	.421	11.8	.268	.294	.279	20.4	.174	.196	.183
2.4	.763	.780	.771	6.7	.405	.432	.417	12.0	.264	.290	.275	20.6	.173	.194	.182
2.5	.748	.766	.756	6.8	.401	.428	.413	12.2	.261	.287	.272	20.8	.171	.193	.180
2.6	.732	.750	.740	6.9	.396	.424	.408	12.4	.257	.283	.268	21.0	.169	.192	.178
2.7	.718	.736	.726	7.0	.393	.421	.405	12.6	.254	.280	.265	21.2	.168	.191	.177
2.8	.705	.723	.713	7.1	.389	.417	.401	12.8	.251	.277	.262	21.4	.167	.190	.176
2.9	.692	.710	.700	7.2	.385	.413	.397	13.0	.248	.274	.259	21.6	.165	.188	.174
3.0	.680	.699	.688	7.3	.381	.410	.393	13.2	.245	.271	.256	21.8	.164	.187	.173
3.1	.668	.687	.676	7.4	.377	.406	.390	13.4	.242	.268	.253	22.0	.163	.186	.172
3.2	.656	.675	.664	7.5	.373	.402	.386	13.6	.239	.265	.250	22.2	.162	.185	.171
3.3	.645	.664	.653	7.6	.370	.399	.383	13.8	.236	.262	.247	22.4	.161	.184	.170
3.4	.634	.653	.642	7.7	.367	.396	.380	14.0	.234	.260	.245	22.6	.160	.183	.169
3.5	.622	.642	.631	7.8	.363	.392	.376	14.2	.231	.257	.242	22.8	.159	.182	.168
3.6	.612	.632	.621	7.9	.360	.389	.373	14.4	.228	.254	.239	23.0	.158	.180	.167
3.7	.602	.622	.611	8.0	.356	.385	.370	14.6	.225	.251	.236	23.2	.156	.179	.165
3.8	.593	.613	.602	8.1	.353	.382	.367	14.8	.223	.249	.234	23.4	.155	.178	.164
3.9	.584	.604	.593	8.2	.350	.379	.364	15.0	.221	.247	.232	23.6	.154	.177	.163
4.0	.572	.596	.583	8.3	.347	.376	.361	15.2	.219	.245	.230	23.8	.153	.176	.162
4.1	.565	.587	.575	8.4	.344	.373	.358	15.4	.217	.242	.227	24.0	.151	.174	.160
4.2	.556	.578	.566	8.5	.341	.371	.355	15.6	.215	.240	.225	24.2	.150	.173	.159
4.3	.548	.570	.558	8.6	.338	.368	.352	15.8	.213	.238	.223	24.4	.149	.172	.158
4.4	.540	.563	.550	8.7	.335	.364	.349	16.0	.211	.236	.221	24.6	.148	.171	.157
4.5	.532	.555	.542	8.8	.332	.361	.346	16.2	.209	.234	.219	24.8	.147	.170	.156
4.6	.525	.548	.535	8.9	.330	.358	.34	16.4	.207	.232	.217	25.0	.146	.169	.155
4.7	.518	.542	.528	9.0	.327	.355	.340	16.6	.205	.230	.215				
4.8	.511	.535	.521	9.1	.324	.353	.337	16.8	.203	.228	.213				
4.9	.504	.528	.514	9.2	.322	.351	.335	17.0	.201	.226	.211				
5.0	.496	.522	.506	9.3	.320	.348	.332	17.2	.199	.224	.209				
5.1	.490	.515	.500	9.4	.317	.345	.329	17.4	.197	.222	.207				
5.2	.484	.509	.494	9.5	.315	.343	.327	17.6	.195	.220	.205				

Column r, the ratio of expansion $= \frac{v_2}{v_1}$

" A, ratio of mean to initial pressure, $\frac{p_m}{p_1} = \frac{10 - 9r^{-\frac{1}{9}}}{r}$ { For dry steam, expanded without gain or loss of heat, in a non-conducting cylinder.

" B, " " " " $\frac{p_m}{p_1} = \frac{1 + \text{hyp. log } r}{r}$ { For damp steam, expanded receiving heat.

" C, " " " " $\frac{p_m}{p_1} = \frac{17 - 16r^{-\frac{1}{16}}}{r}$ { For dry steam, expanded receiving heat sufficient to prevent liquefaction.

RULE – To find the mean pressure exerted throughout the stroke, multiply the initial pressure by the number opposite the ratio of expansion, in the column corresponding with the conditions of expansion.

IV.

TERMINAL PRESSURE RATIOS $\frac{p_1}{p_2}$

r	A	B	C	r	A	B	C	r	A	B	C	r	A	B	C
1.0	0.00	0.0	0.00	4.7	5.58	4.7	5.18	8.3	10.5	8.3	9.47	13.8	18.5	13.8	16.2
1.1	1.11	1.1	1.11	4.8	5.70	4.8	5.29	8.4	10.6	8.4	9.59	14.0	18.8	14.0	16.5
1.2	1.22	1.2	1.21	4.9	5.84	4.9	5.41	8.5	10.7	8.5	9.64	14.2	19.1	14.2	16.8
1.3	1.34	1.3	1.32	5.0	5.98	5.0	5.52	8.6	10.9	8.6	9.76	14.4	19.4	14.4	17.0
1.4	1.45	1.4	1.43	5.1	6.11	5.1	5.64	8.7	11.0	8.7	9.88	14.6	19.7	14.6	17.2
1.5	1.57	1.5	1.54	5.2	6.24	5.2	5.76	8.8	11.2	8.8	10.0	14.8	20.0	14.8	17.5
1.6	1.69	1.6	1.65	5.3	6.38	5.3	5.88	8.9	11.3	8.9	10.2	15.0	20.3	15.0	17.8
1.7	1.80	1.7	1.75	5.4	6.51	5.4	6.00	9.0	11.5	9.0	10.3	15.2	20.6	15.2	18.0
1.8	1.92	1.8	1.87	5.5	6.64	5.5	6.12	9.1	11.6	9.1	10.4	15.4	20.9	15.4	18.2
1.9	2.04	1.9	1.98	5.6	6.78	5.6	6.23	9.2	11.8	9.2	10.6	15.6	21.2	15.6	18.5
2.0	2.16	2.0	2.08	5.7	6.91	5.7	6.35	9.3	11.9	9.3	10.7	15.8	21.5	15.8	18.7
2.1	2.28	2.1	2.20	5.8	7.05	5.8	6.47	9.4	12.0	9.4	10.8	16.0	21.8	16.0	19.0
2.2	2.40	2.2	2.31	5.9	7.18	5.9	6.59	9.5	12.2	9.5	10.9	16.2	22.1	16.2	19.3
2.3	2.52	2.3	2.42	6.0	7.32	6.0	6.71	9.6	12.3	9.6	11.0	16.4	22.4	16.4	19.5
2.4	2.64	2.4	2.53	6.1	7.45	6.1	6.83	9.7	12.5	9.7	11.1	16.6	22.7	16.6	19.8
2.5	2.76	2.5	2.64	6.2	7.59	6.2	6.95	9.8	12.6	9.8	11.3	16.8	23.0	16.8	20.0
2.6	2.89	2.6	2.76	6.3	7.73	6.3	7.07	9.9	12.8	9.9	11.4	17.0	23.3	17.0	20.3
2.7	3.01	2.7	2.87	6.4	7.86	6.4	7.18	10.0	12.9	10.0	11.5	17.2	23.6	17.2	20.5
2.8	3.14	2.8	2.99	6.5	8.00	6.5	7.30	10.2	13.2	10.2	11.7	17.4	23.9	17.4	20.8
2.9	3.26	2.9	3.10	6.6	8.14	6.6	7.42	10.4	13.5	10.4	12.0	17.6	24.2	17.6	21.0
3.0	3.39	3.0	3.21	6.7	8.27	6.7	7.54	10.6	13.8	10.6	12.3	17.8	24.5	17.8	21.3
3.1	3.51	3.1	3.32	6.8	8.41	6.8	7.66	10.8	14.1	10.8	12.5	18.0	24.8	18.0	21.6
3.2	3.64	3.2	3.43	6.9	8.55	6.9	7.78	11.0	14.3	11.0	12.8	18.2	25.1	18.2	21.8
3.3	3.77	3.3	3.55	7.0	8.69	7.0	7.90	11.2	14.6	11.2	13.0	18.4	25.4	18.4	22.0
3.4	3.89	3.4	3.67	7.1	8.83	7.1	8.02	11.4	14.9	11.4	13.3	18.6	25.7	18.6	22.3
3.5	4.02	3.5	3.79	7.2	8.96	7.2	8.14	11.6	15.2	11.6	13.5	18.8	26.0	18.8	22.5
3.6	4.15	3.6	3.90	7.3	9.10	7.3	8.27	11.8	15.5	11.8	13.7	19.0	26.3	19.0	22.8
3.7	4.28	3.7	4.01	7.4	9.24	7.4	8.38	12.0	15.8	12.0	14.0	19.2	26.6	19.2	23.1
3.8	4.41	3.8	4.13	7.5	9.38	7.5	8.49	12.2	16.1	12.2	14.2	19.4	26.9	19.4	23.3
3.9	4.54	3.9	4.25	7.6	9.52	7.6	8.62	12.4	16.4	12.4	14.5	19.6	27.2	19.6	23.6
4.0	4.66	4.0	4.36	7.7	9.66	7.7	8.74	12.6	16.7	12.6	14.8	19.8	27.5	19.8	23.9
4.1	4.79	4.1	4.47	7.8	9.80	7.8	8.87	12.8	17.0	12.8	15.0	20.0	27.9	20.0	24.1
4.2	4.92	4.2	4.60	7.9	9.94	7.9	8.99	13.0	17.3	13.0	15.2	21.0	29.5	21.0	25.4
4.3	5.05	4.3	4.72	8.0	10.1	8.0	9.11	13.2	17.6	13.2	15.5	22.0	31.0	22.0	26.7
4.4	5.18	4.4	4.84	8.1	10.2	8.1	9.23	13.4	17.9	13.4	15.7	23.0	32.6	23.0	28.0
4.5	5.32	4.5	4.95	8.2	10.3	8.2	9.35	13.6	18.2	13.6	16.0	24.0	34.1	24.0	29.3
4.6	5.45	4.6	5.06												

Column r, ratio of expansion $= \frac{v_2}{v_1}$

" A, ratio of initial to final pressure, $p_2 = \frac{p_1}{r^{\frac{10}{9}}}$ .. For dry steam, expanded without gain or loss of heat in a non-conducting cylinder.

" B, " " " " " $p_2 = \frac{p_1}{r}$ .. For damp steam, expanded receiving heat

" C, " " " " " $p_2 = \frac{p_1}{r^{\frac{17}{16}}}$ For dry steam, expanded receiving sufficient heat to prevent liquefaction.

RULE.—To find the final pressure obtaining with any ratio of expansion, divide the initial pressure by the number opposite the ratio of expansion, in the column corresponding with the conditions of expansion.

V.

WORKING OF STEAM.—(NORTHCOTT.)

HEAT-TRANSFER AND TRANSFORMATION.

Initial absolute pressure per sq. in.	Ratio of Expansion	Mean total pressure.	Mean Effective Pressure.	Mean Back Pressure.	Pressure at Release.	Indicated Work per lb. of Steam.	Steam per Indicated Horse-power per Hour	Piston Displacement per lb of Steam	Piston Displacement per Indicated Horse power per Hour	Piston Area per Indicated Horse power with speed of 330 ft per minute	Heat entering Cylinder per lb of Steam	Heat imparted during Expansion per lb of Steam	Heat expended per lb. of Steam	Heat converted into Motive Power indicated per lb of Steam	Heat carried off with the Exhaust Steam per lb	Heat expended per Indicated Horse-power per Hour	Efficiency of Steam.	Coal per Indicated Horse-power per Hour with Boiler of 71 efficiency.
p_1	r	p_m	p_e	p_b	p_r	Ft lbs	Lbs	Cu ft.	Cu. ft.	Sq in	Units	Units	Units	Units	Units	Units.	E.	Lbs.
Class 1											From 212° F		From 212° F		From 212° F	From 212° F		
60	1	60	44	16	60	44,660	44 4	7 0500	311 0	2 37	991	0	991	57 8	933 2	44,001	.0580	4 40
80	1	80	64	16	80	49 698	39 9	5 3926	215 0	1 56	997	0	997	64 4	932 6	39 781	.0644	3 98
100	1	100	84	16	100	52,952	37 4	4 3777	163 7	1 19	1,002	0	1,002	68 6	933 4	37 475	.0684	3 73
120	1	120	104	16	120	55,234	35 8	3 6887	132 0	0 96	1,006	0	1,006	71 5	934 5	35,015	.0710	3 61
150	1	150	134	16	150	57 671	34 4	2 9888	102 8	0 75	1,011	0	1,011	74 7	936 3	34,770	.0737	3 48
200	1	200	184	16	200	60,475	32 7	2 2814	74 6	0 54	1,018	0	1,018	78 3	939 7	33,289	.0770	3 33
250	1	250	234	16	250	62,454	31 7	1 8513	58 7	0 43	1,024	0	1,024	80 9	943 1	32,461	.0790	3 25
300	1	300	284	16	300	64,016	30 9	1 5653	48 3	0 35	1,029	0	1,029	82 9	946 1	31,797	.0806	3 18
Class 2											From 212° F		From 212° F		From 212° F	From 212° F		
60	2	50 7	34 7	16	30	70,445	29 5	14 1000	396 2	2 89	991	0	991	91 3	899 7	29,247	.0877	2 93
80	2	67 6	51 6	16	40	80,139	25 9	10 7850	280 4	1 94	997	0	997	103 8	893 2	25 825	.1031	2 59
100	2	84 6	68 6	16	50	86 490	24 0	8 7554	210 3	1 46	1,002	0	1,002	112 0	890 0	24,049	.1066	2 41
120	2	101 5	85 5	16	60	90 840	22 9	7 3774	168 8	1 17	1,006	0	1,006	117 6	888 4	23,038	.1103	2 31
150	2	126 9	110 9	16	75	95,442	21 8	5 9770	124 1	0 90	1,011	0	1,011	123 6	887 4	22,040	.1118	2 21
200	2	169 2	153 2	16	100	[illegible]	20 7	4 5648	89 9	0 65	1,018	0	1,018	130 3	887 7	21,071	.1127	2 11
250	2	211 5	195 5	16	125	104,406	19 9	3 7066	70 4	0 51	1,024	0	1,024	135 2	888 8	20,478	.1152	2 09
300	2	253 8	237 8	16	150	107,202	19 4	3 1306	57 9	0 42	1,029	0	1,029	138 8	890 2	19,963	.1180	1 99

Class 3											From 212° F.		From 212° F.		From 212° F.	From 212° F		
60	3.4	38.0	22.0	16	16	77,200	25.7	24.32	625.0	4.54	991	77	1,068	100	968	27,448	.0934	2.75
80	4.5	43.1	27.1	16	16	94,956	20.9	24.32	508.3	3.69	997	94	1,091	123	968	22,802	.1120	2.28
100	5.5	47.1	31.1	16	16	108,852	18.2	24.32	441.6	3.21	1,002	107	1,109	141	968	20,184	.1270	2.02
120	6.6	50.6	34.6	16	16	121,204	16.3	24.32	396.4	2.89	1,006	119	1,125	157	968	18,338	.1398	1.84
150	8.1	54.1	38.1	16	16	133,556	14.8	24.32	359.9	2.61	1,011	130	1,141	173	968	16,887	.1518	1.69
250	10.6	59.6	43.6	16	16	152,856	12.9	24.32	313.7	2.28	1,018	148	1,166	198	968	15,042	.1705	1.51
200	13.1	65.8	47.8	16	16	167,524	11.8	24.32	287.0	2.08	1,024	161	1,185	217	968	13,984	.1834	1.40
300	15.5	67.1	51.1	16	16	179,104	11.0	24.32	267.5	1.94	1,029	171	1,200	232	968	13,200	.1943	1.32
Class 4											From 102° F.		From 102° F		From 102° F	From 102° F		
60	3.4	38.0	35.0	3	16	122,826	16.1	24.32	391.5	2.84	1,101	77	1,178	159.0	1,019	18,966	.1352	1.90
80	4.5	43.1	40.1	3	16	140,582	14.0	24.32	330.4	2.40	1,107	94	1,201	182.0	1,019	16,815	.1465	1.69
100	5.5	47.1	44.1	3	16	154,478	12.8	24.32	311.2	2.26	1,112	107	1,219	200.0	1,019	15,604	.1624	1.57
120	6.6	50.6	47.6	3	16	166,830	11.8	24.32	286.9	2.08	1,116	119	1,235	216.0	1,019	14,573	.1739	1.46
150	8.1	54.1	51.1	3	16	179,182	11.0	24.32	267.5	1.94	1,121	130	1,251	232.0	1,019	13,761	.1856	1.38
200	10.7	59.6	56.6	3	16	198,482	9.9	24.32	240.7	1.75	1,128	148	1,276	257.0	1,019	12,633	.2000	1.27
250	13.1	65.8	62.8	3	16	213,150	9.2	24.32	223.7	1.62	1,134	161	1,295	276.0	1,019	11,914	.2132	1.20
300	15.5	67.1	64.1	3	16	224,730	8.8	24.32	214.0	1.55	1,139	171	1,310	291.0	1,019	11,528	.2138	1.16
Classes 5 & 6											From 102° F.		From 102° F		From 102° F.	From 102° F		
60	16.7	12.8	9.8	3	3	166,752	11.9	118	1,404	10.2	1,101	170	1,271	216	1,055	15,125	.1695	1.52
80	23.7	13.8	10.8	3	3	184,508	10.7	118	1,262	9.2	1,107	184	1,294	239	1,055	13,846	.1852	1.39
100	26.9	14.7	11.7	3	3	198,404	10.0	118	1,180	8.5	1,112	200	1,312	257	1,055	13,120	.1954	1.32
120	32.0	15.4	12.4	3	3	210,756	9.4	118	1,109	8.0	1,116	212	1,328	273	1,055	12,484	.2052	1.25
150	39.5	16.2	13.2	3	3	223,108	8.6	118	1,050	7.6	1,121	223	1,344	289	1,055	11,961	.2144	1.20
200	51.7	17.3	14.3	3	3	242,408	8.2	118	968	7.0	1,128	241	1,369	314	1,055	11,226	.2284	1.13
250	63.6	18.2	15.2	3	3	257,076	7.7	118	900	6.6	1,134	254	1,384	333	1,055	10,688	.2399	1.07
300	75.3	18.8	15.8	3	3	268,656	7.4	118	873	6.1	1,139	264	1,403	348	1,055	10,383	.2470	1.04

Class No.	Kind of Engine.
1	Non-condensing; $r = 1$.
2	" " $r = 2$.
3	" " expanding to back-pressure.
4	Condensing; moderate expansion.
5	" jacketed, full expansion.
6	" jacketed, compound expansion complete.

VI.

COMPARISON OF THERMOMETERS.

Celsius	Réaumur.	Fahrenheit.	Celsius.	Réaumur.	Fahrenheit.	Celsius.	Réaumur.	Fahrenheit.
−20	−16	−4	25	20.0	77.0	70	56.0	158.0
−19	−15.2	−2.2	26	20.8	78.8	71	56.8	159.8
−18	−14.4	−0.4	27	21.6	80.[illegible]	72	57.[illegible]	161.6
−17	−13.6	1.4	28	22.4	82.4	73	58.[illegible]	163.4
−16	−12.8	3.2	29	23.2	84.2	74	59.2	165.2
−15	−12.0	5.0	30	24.0	86.0	75	60.0	167.0
−14	−11.2	6.8	31	24.[illegible]	87.8	76	60.8	168.8
−13	−10.4	8.6	32	25.6	89.6	77	61.6	170.6
−12	−9.6	10.4	33	[illegible].4	91.4	78	62.4	172.4
−11	−8.8	12.2	34	27.2	[illegible].2	79	63.2	174.2
−10	−8.0	14.0	35	28.0	95.0	80	64 0	176.0
−9	−7.2	15.8	36	28.8	96.8	81	64.8	177.8
−8	−6.4	17 6	37	29.6	98.6	82	65.6	179.6
−7	−5.6	19.4	38	30.4	100.4	83	66.4	181.4
−6	−4.8	21.2	39	31.2	102.2	84	67.2	183.2
−5	−4.0	23.0	40	32.0	104.0	85	[illegible].0	185.0
−4	−3.2	24.8	41	32.[illegible]	105.8	86	68.8	186.8
−3	−2.4	26.6	42	33.6	107.6	87	69.6	188 6
−2	−1.6	28.4	43	34.4	109.4	88	70.4	190.4
−1	−0.8	30.2	44	35 2	111 2	89	71.2	192.2
0	0	32.0	45	36.0	113.0	90	72.0	194.0
1	0.8	33.8	46	36.8	114.8	91	72.8	195.8
2	1.6	35.6	47	37.6	116.6	92	73.6	197.6
3	2.4	37.4	48	38.4	118.4	93	74.4	199.4
4	3.2	39.2	49	39.2	120.2	94	75.2	201.2
5	4.0	41.0	50	40.0	122.0	95	76.0	203.0
6	4.8	42.8	51	40 8	123.8	96	76.8	204.8
7	5.6	44.6	52	41.6	125.6	97	77.6	206.6
8	6.4	46.4	53	42.4	127.4	98	78.4	208.4
9	7.2	48.2	[illegible]	43.2	129.2	99	79.2	210.2
10	8.0	50.0	[illegible]	44.0	131.0	100	80.0	212.0
11	8.8	51.8	56	44.8	132.8	101	80.8	213.8
12	9.6	53.6	57	[illegible].6	134.6	102	81.6	215.6
13	10.4	55 4	58	46.4	136.4	103	82.4	217.4
14	11.2	57.2	59	47.2	138.2	104	83 2	219.2
15	12.0	59 0	60	48.0	140.0	105	84.0	221.0
16	12.8	60.8	61	48.8	141.8	106	84.8	222.8
17	13 6	62.6	62	49.6	143.6	107	85.6	224.6
18	14.4	64.4	63	50.4	145.4	108	86.4	[illegible].4
19	15.2	66.2	64	51.2	147.2	109	87 2	228.2
20	16.0	68 0	65	52.0	149 0	110	88.0	230.0
21	16.8	69.8	66	52.8	150.8	111	88.8	231.8
22	17.6	71.6	67	53.6	152 6	112	89.6	233.6
23	18.4	73.4	68	54.4	154 4	113	90.4	235.4
24	19.2	75.2	69	55.2	156 2	114	91.2	237.2

COMPARISON OF THERMOMETERS—*Continued.*

Celsius.	Réaumur.	Fahrenheit.	Celsius.	Réaumur.	Fahrenheit.	Celsius.	Réaumur.	Fahrenheit.
115	92.0	239.0	127	101.6	260.6	139	111.2	282.2
116	92.8	240.8	128	102.4	262.4	140	112.0	284.0
117	93.6	242.6	129	103.2	264.2	141	112.8	285.8
118	94.4	244.4	130	104.0	266.0	142	113.6	287.6
119	95.2	246.2	131	104.8	267.8	143	114.4	289.4
120	96.0	248.0	132	105.6	269.6	144	115.2	291.2
121	96.8	249.8	133	106.4	271.4	145	116.0	293.0
122	97.6	251.6	134	107.2	273.2	146	116.8	294.8
123	98.4	253.4	135	108.0	275.0	147	117.6	296.6
124	99.2	255.2	136	108.8	276.8	148	118.4	298.4
125	100.0	257.0	137	109.6	278.6	149	119.2	300.2
126	100.8	258.8	138	110.4	280.4	150	120.0	302.0

VII.

DENSITIES AND VOLUMES OF WATER.

KOPP; CORRECTED BY PORTER.

Temperature. F.	Temperature. C.	Volume, Kopp.	Corrected Volume.	Differences.	Differences.
39.2	4	1.00000	1.00000		
41.0	5	1.00001	1.00001		
51.8	10	1.00025	1.00025	24	34
59.0	15	1.00082	1.00083	58	30
68.0	20	1.00169	1.00171	88	27
77.0	25	1.00284	1.00286	115	24
86.0	30	1.00423	1.00425	139	22
95.0	35	1.00583	1.00586	161	20
104.0	40	1.00768	1.00767	181	19
113.0	[redacted]	1.00967	1.00967	200	19
122.0	50	1.01190	1.01186	219	18
131.0	55	1.01423	1.01423	237	18
140.0	60	1.01672	1.01678	255	18
149.0	65	1.01943	1.01951	273	17
150.0	70	1.02238	1.02241	290	17
167.0	75	1.02554	1.02548	307	17
176.0	80	1.02871	1.02872	324	17
185.0	85	1.03202	1.03213	341	16
194.0	90	1.03553	1.03570	357	16
203.0	95	1.03921	1.03943	373	16
212.0	100	1.04312	1.04332	389	

WEIGHTS AND VOLUMES.

Temperature.	Ratio of volume to that of equal weight at maximum density.	Weight of a cubic foot.	Temperature.	Ratio of volume to that of equal weight at maximum density.	Weight of a cubic foot.	Temperature.	Ratio of volume to that of equal weight at maximum density.	Weight of a cubic foot.
Fahr.		Lbs.	Fahr.		Lbs.	Fahr.		Lbs.
32°	1.000129	62.417	210.°	1.04226	59.894	390.°	1.15538	54.030
39.1	1.000000	62.425	212.	1.04312	59.707	400.	1.16366	53.635
40.	1.000004	62.423	220.	1.04668	59.641	410.	1.17218	53.255
50.	1.000253	62.409	230.	1.05142	59.372	420.	1.18090	52.862
60.	1.000929	62.367	240.	1.05633	59.096	430.	1.18982	52.466
70.	1.001981	62.302	250.	1.06144	58.812	440.	1.19898	52.065
80.	1.00332	62.218	260.	1.06679	58.517	450	1.20833	51.662
90.	1.00492	62.119	270	1.07233	58.214	460.	1.21790	51.256
100.	1.00686	62.000	280.	1.07809	57.903	470	1.22767	50.848
110.	1.00902	61.867	290	1.08405	57.585	480	1.23766	50.438
120.	1.01143	61.720	300.	1.09023	57.259	490.	1.24785	50.026
130.	1.01411	61.556	310.	1.09661	56.925	500.	1.25828	49.611
140.	1.01690	61.388	320.	1.10323	56.584	510.	1.26892	49.195
150	1.01995	61.204	330.	1.11005	56.236	520.	1.27975	48.778
160	1.02324	61.007	340.	1.11706	55.883	530.	1.29080	48.360
170	1.02671	60.801	350.	1.12431	55.523	540.	1.30204	47.941
180	1.03033	60.587	360	1.13175	55.158	550.	1.31354	47.521
190.	1.03411	60.366	370.	1.13942	54.787			
200.	1.03807	60.136	380.	1.14729	54.411			

VIII.

TEMPERATURES AND PRESSURES, SATURATED STEAM.

IN METRIC MEASURES AND FROM REGNAULT.

Temperature.	Steam-pressure.		Temperature.	Steam-pressure.	
	In Centimetres.	In Atmospheres		In Centimetres.	In Atmospheres
− 32° C.	0.0330	0.0004	+ 14° C.	1.1908	0.016
31	0.0352	0.0005	15	1.2699	0.017
30	0.0386	0.0005	16	1.3536	0.018
29	0.0424	0.0006	17	1.4421	0.019
28	0.0464	0.0006	18	1.5357	0.020
27	0.0508	0.0007	19	1.6346	0.022
26	0.0555	0.0007	20	1.7391	0.023
25	0.0605	0.0008	21	1.8495	0.024
24	0.0660	0.0009	22	1.9659	0.026
23	0.0719	0.0009	23	2.0888	0.028
22	0.0783	0.0010	24	2.2184	0.029
21	0.0853	0.0011	25	2.3550	0.031
20	0.0927	0.0012	26	2.4988	0.033
19	0.1008	0.0013	27	2.5505	0.034
18	0.1095	0.0014	28	2.8101	0.037
17	0.1189	0.0015	29	2.9782	0.039
16	0.1290	0.0017	30	3.1548	0.042
15	0.1400	0.0018	31	3.3406	0.044
14	0.1518	0.0020	32	3.5359	0.047
13	0.1646	0.0022	33	3.7411	0.049
12	0.1783	0.0024	34	3.9565	0.052
11	0.1933	0.0025	35	4.1827	0.055
10	0.2093	0.0027	36	4.4201	0.058
9	0.2267	0.0030	37	4.6691	0.061
8	0.2455	0.0032	38	4.9302	0.065
7	0.2658	0.0035	39	5.2039	0.068
6	0.2876	0.0038	40	5.4906	0.072
5	0.3113	0.0041	41	5.7910	0.076
4	0.3368	0.0044	42	6.1055	0.080
3	0.3644	0.0048	43	6.4346	0.085
2	0.3941	0.0052	44	6.7790	0.089
1	0.4263	0.0056	45	7.1391	0.094
0	0.4600	0.0061	46	7.5158	0.099
+ 1	0.4940	0.0065	47	7.9093	0.104
2	0.5302	0.0070	48	8.3204	0.109
3	0.5687	0.0073	49	8.7499	0.115
4	0.6097	0.0080	50	9.1982	0.121
5	0.6534	0.0086	51	9.6661	0.127
6	0.6998	0.0092	52	10.1543	0.134
7	0.7492	0.0199	53	10.6636	0.140
8	0.8017	0.0107	54	11.1945	0.147
9	0.8574	0.011	55	11.7478	0.155
10	0.9165	0.012	56	12.3244	0.163
11	0.9792	0.013	57	12.9251	0.170
12	1.0457	0.014	58	13.5505	0.178
13	1.1162	0.015	59	14.2015	0.187

Temperatures and Pressures, Saturated Steam—*Continued.*

Temperature.	Steam-pressure. In Centimetres.	Steam-pressure. In Atmospheres.	Temperature.	Steam-pressure. In Centimetres.	Steam-pressure. In Atmospheres.
+ 60° C.	14.8791	0.196	+110 C	107 537	1.415
61	15.5839	0.205	111	111.209	1.463
62	16 3170	0.215	112	114 983	1.513
63	17.0791	0 225	113	118.861	1.564
64	17.8714	0.235	114	122 847	1.616
65	18.6945	0.246	115	126.941	1.670
66	19 5496	0.257	116	131.147	1.726
67	20 4376	0.267	117	135.466	1.782
68	21.3596	0.281	118	139.902	1.841
69	22.3165	0 294	119	144.455	1.901
70	23 3093	0.306	120	149.128	1.962
71	24 3393	0.320	121	153.925	2.025
72	25 4073	0.334	122	158.847	2 091
73	26 5147	0 349	123	163.896	2.157
74	27 6624	0.364	124	169 076	2.225
75	28 8517	0.380	125	174 388	2.295
76	30 0838	0.396	126	179.835	2 366
77	31 3600	0.414	127	185.420	2.430
78	32 6811	0.430	128	191.147	2.515
79	34 0488	0.448	129	197.015	2.592
80	35 4643	0.466	130	203 028	2.671
81	36.9287	0.486	131	209 194	2.753
82	38 4435	0.506	132	215.503	2 836
83	40 0101	0.526	133	221.969	2.921
84	41.6298	0 548	134	228.592	3 008
85	43 3041	0.570	135	235.373	3.097
86	45 0344	0.593	136	242.316	3.188
87	46.8221	0 616	137	249.423	3.282
88	48 6687	0.640	138	256.700	3.378
89	50 5759	0.665	139	264.144	3.476
90	52 5450	0.691	140	271.763	3.576
91	54 5778	0.719	141	279 557	3.678
92	56 6757	0 746	142	287 530	3.783
93	58 8406	0 774	143	295 686	3.890
94	61 0740	0.804	144	304 026	4.000
95	63.3778	0.834	145	312.555	4.113
96	65 7535	0.865	146	321 274	4.227
97	68.2029	0.897	147	330.187	4.344
98	70.7280	0 931	148	339 298	4 464
99	73 3305	0.965	149	348 609	4.587
100	76 000	1 000	150	358.123	4.712
101	76 7590	1.036	151	367 843	4.840
102	81 6010	1.074	152	377 774	4.971
103	84 5280	1.112	153	387 918	5.104
104	87 5410	1.152	154	398.277	5.240
105	90 6410	1 193	155	408 856	5.380
106	93 8310	1 235	156	419.659	5.522
107	97 1140	1 278	157	430.688	5 667
108	100 4910	1 322	158	441.945	5.815
109	103 965	1 368	159	453 436	5.966

TEMPERATURES AND PRESSURES, SATURATED STEAM—*Continued.*

Temperature.	Steam-pressure. In Centimetres.	Steam-pressure. In Atmospheres	Temperature.	Steam-pressure. In Centimetres.	Steam-pressure. In Atmospheres
+160° C.	465.162	6.120	+196° C.	1074.595	14.199
161	477.125	6.278	197	1097.500	14.441
162	489.336	6.439	198	1120.962	14.749
163	501.791	6.603	199	1144.746	15.062
164	514.497	6.770	200	1168.896	15.380
165	527.454	6.940	201	1193.457	15.703
166	540.669	7.114	202	1218.369	16.031
167	554.143	7.291	203	1243.700	16.364
168	567.882	7.478	204	1269.490	16.703
169	581.890	7.656	205	1295.566	17.047
170	596.166	7.844	206	1322.112	17.396
171	610.719	8.036	207	1349.075	17.751
172	625.543	8.231	208	1376.453	18.111
173	640.660	8.430	209	1404.252	18.477
174	656.055	8.632	210	1432.480	18.848
175	671.743	8.839	211	1461.132	19.226
176	687.722	9.049	212	1490.222	19.608
177	703.997	9.263	213	1519.748	19.997
178	720.572	9.481	214	1549.717	20.391
179	737.452	9.703	215	1580.133	20.791
180	754.639	9.929	216	1610.994	21.197
181	772.137	10.150	217	1642.315	21.690
182	789.952	10.394	218	1674.090	22.027
183	808.084	10.633	219	1706.329	22.452
184	826.540	10.876	220	1739.036	22.882
185	845.323	11.123	221	1772.213	23.319
186	864.435	11.374	222	1805.864	23.761
187	883.882	11.630	223	1839.994	24.210
188	903.668	11.885	224	1874.607	24.666
189	923.795	12.155	225	1909.704	25.128
190	944.270	12.425	226	1945.292	25.596
191	965.093	12.699	227	1981.376	26.071
192	986.271	12.977	228	2017.961	26.552
193	1007.804	13.261	229	2055.048	27.040
194	1029.701	13.549	230	2092.640	27.535
195	1051.963	13.842			

IX.

METRIC STEAM AND WORK TABLE.

Absolute pressures in Atmosphere.	Specific volumes v_0 in Cu. meters.	Product p_0v_0.	$W = \frac{26127.34}{1000\ p_0v_0}$	$W \cdot p_0$.
0.1	14.504	1.450	18.010	1.801
0.2	7.525	1.505	17.418	3.483
0.3	5.128	1.540	16.960	5.088
0.4	3.908	1.560	16.750	6.700
0.5	3.165	1.580	16.530	8.265
0.6	2.665	1.600	16.339	9.803
0.7	2.304	1.610	16.230	11.361
0.8	2.031	1.620	16.120	12.896
0.9	1.818	1.630	16.020	14.418
1.0	1.646	1.646	15.870	15.870
1.1	1.505	1.655	15.780	17.385
1.2	1.386	1.663	15.710	18.852
1.3	1.285	1.670	15.640	20.332
1.4	1.199	1.680	15.540	21.756
1.5	1.123	1.684	15.510	23.265
1.6	1.057	1.691	15.450	24.720
1.7	0.999	1.699	15.370	26.129
1.8	0.946	1.703	15.340	27.612
1.9	0.899	1.708	15.290	29.051
2.0	0.857	1.714	15.243	30.456
2.1	0.819	1.718	15.208	31.937
2.2	0.784	1.725	15.146	33.321
2.3	0.751	1.727	15.128	34.794
2.4	0.722	1.733	15.076	36.182
2.5	0.695	1.741	15.002	37.505
2.6	0.670	1.742	14.990	38.974
2.7	0.646	1.744	14.970	40.190
2.8	0.625	1.750	14.929	41.801
2.9	0.604	1.752	14.921	43.271
3.0	0.586	1.758	14.861	44.583
3.1	0.568	1.761	14.838	45.998
3.2	0.551	1.765	14.818	47.417
3.3	0.535	1.765	14.700	48.807
3.4	0.521	1.771	14.740	50.146
3.5	0.507	1.774	14.725	51.530
3.6	0.493	1.775	14.720	52.992
3.7	0.481	1.780	14.680	54.316
3.8	0.469	1.782	14.660	55.708
3.9	0.457	1.784	14.650	57.057
4.0	0.447	1.788	14.63	58.440
4.1	0.437	1.792	14.58	59.778
4.2	0.427	1.795	14.56	61.152
4.3	0.418	1.797	14.55	62.479
4.4	0.408	1.799	14.52	65.888

Metric Steam and Work Table—*Continued.*

Absolute pressure p_0 in Atmospheres.	Specific volumes v_0 in Cu. Meters.	Product $p_0 v_0$	$W = \frac{26127.34}{2000\, p_0 v_0}$	$W \cdot p_0$
4.5	0.400	1.800	14.51	65.295
4.6	0.392	1.803	14.49	66.654
4.7	0.384	1.805	14.45	67 915
4.8	0.377	1.810	14 43	69 264
4.9	0 370	1.813	14.41	70 609
5.0	0.363	1 815	14 39	71 950
5.1	0.356	1 810	14.38	73 338
5.2	0 350	1.820	14.36	74 672
5.3	0 343	1.821	14 35	76.055
5.4	0 337	1 823	14.33	77 382
5.5	0 332	1.825	14 31	78.705
5.6	0.326	1 826	14.30	80.080
5.7	0 321	1 829	14.26	81.282
5.8	0.316	1.833	14.25	82.650
5.9	0.311	1.835	14.24	84 016
6.0	0.306	1.836	14.23	85.380
6.25	0.294	1.838	14.21	88.812
6.5	0.284	1.845	14.16	92.040
6.75	0.273	1.848	14.13	95.377
7.0	0.265	1.855	14.10	98.700
7.25	0.256	1.856	14.07	100.997
7.5	0.248	1.860	14.04	105.300
7.75	0.241	1.867	13.99	108.422
8.0	0.234	1.872	13.96	111.680
8.25	0.227	1.873	13.95	114.077
8.5	0.221	1.878	13.91	118 235
8.75	0.215	1.881	13.89	121 537
9.0	0.209	1.883	13.86	124.740
9.25	0.204	1.887	13.84	128 020
9.5	0.199	1.891	13.81	131.195
9.75	0.194	1.893	13.80	134.550
10.0	0.190	1.900	13.75	137.500

X.

PROPERTIES OF SATURATED STEAM.

NOTE.—The following table gives the data required by the engineer in this connection as based upon the experiments of Regnault. The temperatures, pressures, and heat-measures are all from Regnault's experiments. The other quantities were calculated by Mr. R. H. Buel,* adopting the formulas of Rankine already given to obtain quantities not ascertained by direct experiment. The two parts of the latent heat of vaporization are separately determined, and the internal thus distinguished from the external work of expansion. British measures are adopted. The nomenclature is sufficiently well explained by the table headings.

Pressure above a vacuum, in pounds per square inch.	Temperature, Fahrenheit degrees.	Quantities of Heat. In British Thermal Units. Required to raise the temperature of the water from 32° to $T°$.	Internal latent heat.	External latent heat.	Latent heat of evaporation at pressure $P = I + E$.	Total heat of evaporation above 32° $= S + L$.	Total heat of evaporation above 32°, in units of evaporation.	Weight of a cubic foot of steam, in pounds.	Volume. Of a pound of steam in cubic feet.	Ratio of volume of steam to volume of equal weight of distilled water at temperature of maximum density.	Pressure above a vacuum, in pounds per square inch.
P	t	S	I	E	L	H	U	W	C	V	P
1	102 018	70 040	981 306	61 619	1043 015	1113 055	1 1529	.003007	330 4	20,623	1
2	126 302	94 368	961 780	64 114	1026 094	1120 462	1 1599	.005818	171 9	10,730	2
3	141 654	109 764	949 725	65 655	1015 380	1125 144	1 1647	.008522	117 3	7,325	3
4	153 122	121 271	940 597	66 773	1007 370	1128 641	1 1683	.011172	89 51	5,588	4
5	162 370	130 563	933 240	67 660	1000 899	1131 462	1 1719	.013781	72 56	4,530	5
6	170 173	138 401	927 038	68 403	995 441	1133 842	1 1737	.016357	61 14	3,816	6
7	176 945	145 213	921 654	69 041	990 695	1135 908	1 1758	.018908	52 89	3,302	7
8	182 952	151 255	916 883	69 602	986 485	1137 740	1 1777	.021436	46 65	2,912	8
9	188 357	156 699	912 584	70 106	982 690	1139 389	1 1794	.023944	41 77	2,607	9
10	193 284	161 660	908 672	70 560	979 232	1140 892	1 1810	.026437	37 83	2,361	10
11	197 814	166 225	905 084	70 967	976 051	1142 275	1 1824	.028911	34 59	2,159	11
12	202 012	170 457	901 766	71 330	973 096	1143 553	1 1837	.031376	31 87	1,990	12
13	205 929	174 407	898 683	71 663	970 346	1144 748	1 1849	.033828	29 56	1,845	13
14	209 604	178 112	895 784	71 973	967 757	1145 869	1 1861	.036265	27 58	1,721	14
14 69	212 000	180 531	893 894	72 175	966 069	1146 600	1 1869	.037928	26 37	1,646	14 69
15	213 067	181 608	893 044	72 274	965 318	1146 926	1 1872	.038688	25 85	1,614	15
16	216 347	184 919	890 458	72 549	963 007	1147 926	1 1882	.041109	24 33	1,519	16
17	219 452	188 056	888 007	72 811	960 818	1148 874	1 1892	.043519	22 98	1,434	17

* Weisbach's Mechanics, vol. II, part ii. Dubois' translation. N. Y.: J. Wiley & Sons, [illegible]

P	t	S	I	E	L	H	U	W	C	V	P
18	222 424	191 058	885.661	73 060	958 721	1149 779	1.1901	.045900	21 78	1,359	18
19	225.235	193 918	883.427	73 298	956 725	1150 643	1 1910	.048310	20.70	1,292	19
20	227 964	196 655	881 289	73 525	954 814	1151 469	1.1919	.050696	19.73	1,232	20
21	230 565	199 285	879 239	73 739	952 978	1152 269	1 1927	053074	18 84	1,176	21
22	233 069	201 817	877.267	73.942	951 209	1153 026	1 1935	055440	18 04	1,126	22
23	235 479	204 258	875 368	74 136	949 504	1154 762	1 1943	057812	17.30	1,080	23
24	237 803	206 610	873.538	74.323	947 861	1155 471	1 1950	.060171	16 60	1,038	24
25	240.053	208 887	871 767	74.503	946 270	1155 157	1 1957	062504	16 00	998 4	25
26	242 225	211 089	870 032	74 678	944 730	1155 819	1 1964	.064870	15 40	960.3	26
27	244 333	213.223	868 391	74.847	943 238	1156.461	1 1971	.067210	14 [illegible]	[illegible]	27
28	246 376	215 293	866 780	75.011	941 791	1157 084	1 1978	069545	14 38	897 6	28
29	248 363	217 308	865.215	75.168	940 383	1157.691	1 1984	.071875	13 91	868 5	29
30	250 293	219 261	863 700	75 319	939 019	1158 280	1 1990	074201	13 48	841 3	30
31	252 171	221.165	862 221	75.466	937 687	1158 852	1 1996	076520	13 07	815 8	31
32	254 002	223 021	860.781	75.608	936 389	1159 410	1 2002	.078830	12 68	791 8	32
33	255 782	224 827	859 382	75.745	935 127	1159 954	1 2008	.081152	12 31	769 0	33
34	257 523	226 594	858 013	75 878	933 891	1160 485	1 2013	.083461	11 98	748 0	34
35	259 221	228 316	856 680	76 007	932 687	1161 003	1 2018	085766	11 66	727 9	35
36	260 883	230 001	855 375	76 133	931 508	1161 509	1 2023	088067	11 36	708 8	36
37	262.905	231 650	854.099	76 255	930 354	1162 004	1 2028	.090364	11.07	690 8	37
38	264 093	233 261	852.852	76 375	929 227	1162 488	1 2033	.092657	10 79	673 7	38
39	265 647	234 840	851 629	76 493	928 122	1162 962	1 2038	094946	10 53	657 5	39
40	267 168	236 386	850 432	76.608	927 040	1163 426	1 2043	.097231	10 28	642 0	40
41	268 660	237 902	849 261	76 719	925 980	1163 882	1 2048	099514	10 04	627.3	41
42	270 122	239.389	848 113	76.827	924 940	1164 329	1 2053	.101794	9 816	613 3	42
43	271 557	240 846	846 988	76 932	923 920	1164 766	1 2058	104071	9 609	599 9	43
44	272 965	242 275	845.884	77 035	922 919	1165 194	1 2062	106345	9 413	587 0	44
45	274 347	243 680	844 799	77 136	921 935	1165 615	1 2066	108616	9 207	574 7	45
46	275 704	245 061	843 733	77 235	920 968	1166 029	1 2070	110884	9 018	563 0	46
47	277 036	246 418	842.687	77 331	920 018	1166 436	1 2074	113149	8 838	551 7	47
48	278 348	247 752	841 659	77 425	919 084	1166 836	1 2078	.115411	8 664	540 9	48
49	279 637	249 064	840 647	77 517	918 164	1167 228	1 2082	117670	8 498	530 5	49
50	280 904	250 355	839 653	77 607	917 260	1167 615	1 2086	119927	8 338	520 5	50
51	282 151	251 624	838 675	77 696	916 371	1167 995	1 2090	.122181	8 185	510 9	51
52	283 381	252 875	837 700	77 784	915 494	1168 369	1 2094	124433	8 037	501 7	52
53	284 [illegible]	254 106	836 760	77 870	914 632	1168 738	1 2098	126682	7 894	492 8	53
54	285 [illegible]	255 321	835 827	77 954	913 781	1169 102	1 2102	128928	7 756	484 2	54
55	286 955	256 518	834 906	78 036	912 942	1169 460	1 2106	131172	7 624	475 9	55
56	288 111	257 695	834.004	78 117	912 118	1169.813	1 2110	133414	7 496	467 9	56
57	289 251	258 857	833.108	78 196	911 304	1170.162	1 2114	135654	7 370	460.2	57

PROPERTIES OF SATURATED STEAM—(*Continued*).

Pressure above a vacuum, in pounds per square inch.	Temperature, Fahrenheit degrees.	Quantities of Heat. In British Thermal Units. Required to raise the temperature of the water from 32° to $T°$	Internal latent heat.	External latent heat.	Latent heat of evaporation at pressure $P = I + E$	Total heat of evaporation above $32° = S + L$	Total heat of evaporation above 32°, in units of evaporation.	Weight of a cubic foot of steam, in pounds.	Volume. Of a pound of steam in cubic feet.	Ratio of volume of steam to volume of equal weight of distilled water at temperature of maximum density.	Pressure above a vacuum, in pounds per square inch.
P	t	S	I	E	L	H	U	W	C	V	P
58	290.374	260.002	832.228	78.273	910.501	1170.503	1.2117	.137892	7.252	452.7	58
59	291.483	261.132	831.361	78.348	909.709	1170.841	1.2120	.140128	7.136	445.5	59
60	292.575	262.248	830.507	78.421	908.928	1171.176	1.2123	.142362	7.024	438.5	60
61	293.643	263.348	829.663	78.494	908.157	1171.505	1.2127	.144594	6.916	432.7	61
62	294.717	264.433	828.830	78.566	907.396	1171.829	1.2130	.146824	6.811	425.2	62
63	295.768	265.506	828.005	78.638	906.643	1172.149	1.2133	.149052	6.709	418.8	63
64	296.805	266.566	827.191	78.709	905.900	1172.466	1.2136	.151277	6.610	412.6	64
65	297.830	267.612	826.388	78.779	905.167	1172.779	1.2140	.153500	6.515	406.6	65
66	298.842	268.644	825.596	78.847	904.443	1173.087	1.2143	.155721	6.422	400.8	66
67	299.843	269.666	824.814	78.913	903.727	1173.393	1.2146	.157940	6.331	395.2	67
68	300.831	270.674	824.042	78.978	903.020	1173.694	1.2149	.160157	6.244	389.8	68
69	301.807	271.669	823.280	79.042	902.322	1173.991	1.2152	.162372	6.159	384.5	69
70	302.774	272.657	822.524	79.105	901.629	1174.286	1.2155	.164584	6.076	379.3	70
71	303.728	273.633	821.778	79.167	900.945	1174.578	1.2158	.166794	5.995	374.3	71
72	304.669	274.597	821.041	79.228	900.269	1174.866	1.2161	.169003	5.917	369.4	72
73	305.603	275.550	820.312	79.288	899.600	1175.150	1.2164	.171210	5.841	364.6	73
74	306.526	276.493	819.589	79.349	898.938	1175.431	1.2167	.173417	5.767	360.0	74
75	307.440	277.427	818.873	79.410	898.283	1175.710	1.2170	.175622	5.694	355.5	75
76	308.344	278.350	818.166	79.469	897.635	1175.985	1.2173	.177825	5.624	351.1	76

P	t	S	I	E	L	H	U	W	C	V	P
77	309.239	279 265	817 468	79 526	896 994	1176.259	1 2176	.180027	5 555	346 8	77
78	310.123	280 170	816 777	79 582	896 339	1176 529	1 2179	.182229	5 488	342 6	78
79	311.000	281 066	816.090	79.639	845 729	1176 795	1 2181	.184429	5 422	338 5	79
80	311 866	281 952	815 413	79 695	895 108	1177 060	1 2184	186627	5.358	334 5	80
81	312 725	282.830	814 742	79 749	894 491	1177 321	1 2187	.188823	5 296	330 6	81
82	313 576	283.701	814.077	79 802	893 879	1177 580	1 2190	191017	5 235	326.8	82
83	314 417	284 562	813 419	79 856	893 275	1177 837	1 2193	193210	5 176	323 1	83
84	315 250	285 414	812 768	79 909	892 677	1178.091	1 2195	195401	5 118	319.5	84
85	316 076	286 260	812 122	79 961	892 083	1178 343	1 2198	197591	5 061	315.9	85
86	316 893	287 096	811 484	80 012	891 496	1178 592	1 2200	199781	5 006	312.5	86
87	317 705	287.927	810 850	80 063	890 913	1178 840	1 2203	201969	4 951	309.1	87
88	318 510	288.750	810 222	80 113	890 335	1179 085	1 2205	204155	4 898	305 8	88
89	319 306	289 565	809 601	80 162	889 763	1179 328	1 2208	206340	4 846	302 5	89
90	320 094	290.373	808.986	80 210	889 196	1179 569	1 2210	208525	4 796	299.4	90
91	320 877	291 176	808 375	80.258	888 633	1179 809	1 2212	210709	4 746	296.3	91
92	321 653	291 970	807 770	80 305	888 075	1180 045	1 2215	.212892	4 697	293 2	92
93	322 422	292 758	807 170	80 351	887 521	1180 279	1 2217	215074	4 650	290 2	93
94	323 183	293.539	806 575	80 397	886 972	1180 511	1 2220	217253	4 603	287 3	94
95	323 939	294 314	805.985	80.442	886 427	1180 741	1.2222	.219430	4 557	284 5	95
96	324 688	295 083	805 400	80 487	885 887	1180 970	1 2224	221604	4 513	281 7	96
97	325 431	295 845	804 821	80.531	885 352	1181 197	1 2227	223778	4 469	279 0	97
98	326 169	296 601	804 245	80 576	884 821	1181 422	1 2229	.225950	4 426	276 3	98
99	326 900	297 350	803 675	80 620	884 295	1181 645	1 2232	228122	4 384	273 7	99
100	327 625	298 093	803 108	80 665	883.773	1181 866	1 2234	.230293	4 342	271 1	100
101	328 345	298 832	802 544	80 709	883 253	1182 085	1 2236	232464	4.302	268 5	101
102	329 060	299 566	801 985	80 752	882 737	1182 303	1 2238	.234634	4 262	266 0	102
103	329 769	300 293	801 432	80 794	882 226	1182.519	1 2240	236803	4.223	263 6	103
104	330 471	301 014	800 884	80 835	881.719	1182 733	1 2242	238972	4.185	261 2	104
105	331 169	301 731	800 339	80 875	881 214	1182 945	1 2245	.241139	4 147	258 9	105
106	331 862	302 444	799 796	80 916	880 712	1183 156	1 2247	.243304	4.110	256.6	106
107	332 550	303.152	799 258	80 956	880 214	1183 366	1 2249	.245467	4.074	254 3	107
108	333 232	303 854	798 725	80 995	879 720	1183 574	1 2251	.247629	4.038	252 1	108
109	333 911	304 551	798 196	81 034	879 230	1183 781	1 2254	.249789	4.003	249 9	109
110	334 582	305 242	797 672	81 072	878 744	1183 986	1 2256	.251947	3 969	247 8	110
111	335 250	305 927	797 153	81 110	878 263	1184 190	1 2258	.254105	3 935	245 7	111
112	335 914	306 609	796 637	81 147	877 784	1184 393	1 2260	256263	3 902	243 6	112
113	336 573	307 285	796 125	81 184	877 309	1184 594	1 2262	258420	3 870	241 6	113
114	337 226	307 956	795 617	81 221	876 838	1184 794	1 2264	260576	3 838	239 6	114
115	337 874	308 621	795 114	81 257	876 371	1184 992	1 2266	262732	3 806	237 6	115
116	338 518	309 281	794 614	81 293	875 907	1185 188	1 2268	264887	3 775	235 7	116
117	337 159	309 939	794 114	81 330	875 444	1185.383	1 2270	267041	3 745	233.8	117

PROPERTIES OF SATURATED STEAM—(*Continued*).

Pressure above a vacuum, in pounds per square inch.	Temperature, Fahrenheit degrees.	Quantities of Heat. In British Thermal Units. Required to raise the temperature of the water from 32° to T°.	Internal latent heat.	External latent heat.	Latent heat of evaporation at pressure P $= I + E$.	Total heat of evaporation above 32° $= S + L$.	Total heat of evaporation above 32°, in units of evaporation	Weight of a cubic foot of steam, in pounds.	Volume. Of a pound of steam in cubic feet.	Ratio of volume of steam to volume of equal weight of distilled water at temperature of maximum density	Pressure above a vacuum, in pounds per square inch.
P	t	S	I	E	L	H	U	W	C	V	P
118	339 796	310 592	793 619	81 366	874 985	1185 577	1 2272	269195	5 715	231 9	118
119	340 430	311 241	793 126	81 403	874 529	1185 770	1 2274	271348	3 685	230 1	119
120	341 058	311 885	792 637	81 439	874 076	1185 961	1 2276	273500	3 656	228 3	120
121	341 681	312 524	792 152	81 474	873 626	1186 150	1 2278	275651	3 628	226 5	121
122	342 300	313 161	791 669	81 509	873 178	1186 339	1 2280	277801	3 600	224 7	122
123	342 916	313 795	791 189	81 543	872 732	1186 527	1 2282	.279949	3 572	223 0	123
124	343 528	314 425	790.711	81 578	872 289	1186 714	1 2284	282097	3 545	221 3	124
125	344 136	315 051	790 236	81 612	871 848	1186 899	1 2286	284243	3 518	219 6	125
126	344 741	315 672	789 765	81 646	871 411	1187 083	1 2288	286389	3 492	218 0	126
127	345 340	316 289	789 298	81 679	870 977	1187 266	1 2290	288533	3 466	216 4	127
128	345 936	316 903	788 834	81 711	870 545	1187 448	1 2292	290677	3 440	214 8	128
129	346 530	317 513	788 374	81 742	870 116	1187 629	1 2293	292820	3 415	213 2	129
130	347 121	318 121	787 914	81 774	869 688	1187 809	1 2295	294961	3 390	211 6	130
131	347 706	318 725	787 458	81 805	869 264	1187 988	1 2296	297102	3 366	210 1	131
132	348 287	319 325	787 004	81 837	868 841	1188 166	1 2298	299242	3 342	208 6	132
133	348 867	319 922	786 554	81 868	868 422	1188 344	1 2300	301382	3 318	207 1	133
134	349 443	320 515	786 105	81 900	868 005	1188 520	1 2302	303521	3 295	205 7	134
135	350 015	321 105	785 659	81 931	867 590	1188 695	1 2304	305659	3 272	204 2	135
136	350 584	321 690	785 215	81 962	867 177	1188 869	1 2306	307797	3 249	202 8	136
137	351 149	322 274	784 775	81 992	866 767	1189 041	1 2308	309934	3 227	201 4	137
138	351 711	322 853	784 339	82.021	866 360	1189 213	1 2309	312070	3 204	200.0	138

P	t	S	I	E	L	H	U	W	C	V	P
139	352 271	323 429	783.905	82 050	865 955	1189 384	1.2311	.314205	3 182	198 7	139
140	352 827	324 003	783 472	82 080	865 552	1189 555	1 2313	.316338	3 161	197 3	140
141	353 380	324.573	783 042	82 109	865.151	1189 724	1.2315	.318471	3 140	196 0	141
142	353 931	325 141	782 613	82 138	864 751	1189 892	1.2316	.320603	3.119	194 7	142
143	354 478	325 705	782 188	82.166	864 354	1190 059	1.2318	.322735	3.099	193 4	143
144	355 022	326 265	781.766	82 194	863 960	1190.225	1 2320	.324867	3 078	192 2	144
145	355.562	326 823	781 346	82.221	863 567	1190 390	1 2321	.326998	3 058	190 9	145
146	356 100	327.378	780 927	82.249	863 176	1190 554	1 2323	.329128	3.038	189 7	146
147	356 636	327 930	780 510	82 277	862 787	1190 717	1 2324	.331257	3 019	188.5	147
148	357 169	328 479	780.096	82 304	862 400	1190 879	1 2326	.333386	3 000	187.3	148
149	357 697	329.024	779 684	82 332	862 016	1191 040	1 2328	.335515	2 981	186.1	149
150	358 223	329.566	779 275	82.359	861.634	1191.200	1 2330	.337643	2 962	184 9	150
160	363 346	334.850	775 296	82 616	857 912	1192.760	1.2346	.358886	2 786	173 9	160
170	368 226	339 892	771 505	82 854	854 359	1194 251	1 2361	.380071	2 631	164 3	170
180	372 886	344 708	767.891	83.072	850.963	1195 671	1 2376	.401201	2 493	155 6	180
190	377 352	349 329	764.430	83 273	847 703	1197 032	1 2390	.422280	2 368	147 8	190
200	381 636	353 766	761 111	83 462	844.573	1198 339	1 2404	.443310	2 256	140 8	200
210	385 759	358 041	757 916	83 640	841 556	1199 597	1 2417	.464295	2 154	134 5	210
220	389 736	362 168	754 834	83 808	838.642	1200 810	1 2430	.485237	2.061	128.7	220
230	393 575	366 152	751.862	83 966	835 828	1201.980	1 2442	.506139	1 976	123 3	230
240	397 285	370 008	748.988	84 115	833 103	1203 111	1.2454	.527003	1 898	118.5	240
250	400 883	373 750	746 203	84 256	830 459	1204 209	1.2465	.547831	1 825	114 0	250
260	404.370	377.377	743 508	84.388	827 896	1205 273	1 2476	.568626	1.759	109 8	260
270	407 755	380 905	740 891	84 510	825 401	1206 306	1 2487	.589390	1.697	105 9	270
280	411 048	384 337	738.350	84 623	822 973	1207 310	1.2497	.610124	1 639	102.3	280
290	414 250	387 677	735 878	84 731	820 609	1208 286	1 2507	.630829	1 585	99 0	290
300	417 371	390 933	733 470	84 835	818 305	1209 238	1 2517	.651506	1 535	95.8	300
350	431 96	406 26	722 20	85 28	807 48	1213 74	1 256	.754534	1.325	82.7	350
400	444 92	419 76	712 34	85.60	797.94	1217 70	1 260	.857185	1 167	72 8	400
450	456 62	432 18	703 28	85.84	789 12	1221.30	1 264	.959536	1.042	65 1	450
500	467 40	443 52	695 01	86 01	781.02	1224 54	1.267	1.061700	.942	58 8	500
550	477 50	454 14	687 34	86 12	773 46	1227 60	1 270	1 16380	.859	53 6	550
600	486 86	464 22	680 08	86 18	766 26	1230 48	1 273	1 26586	.790	49 3	600
650	495 68	473 58	673.40	86 20	759.60	1233 18	1 276	1 36791	.731	45 6	650
700	504 14	482 40	667 11	86 19	753 30	1235 70	1 279	1.46995	.680	42.4	700
750	512 06	490 86	661 04	86 14	747 18	1238 04	1.282	1 57198	.636	39 6	750
800	519 62	498 88	655 34	86 08	741 42	1240 30	1 285	1.67401	.597	37.1	800
850	526 82	506 66	649.84	86.00	735 84	1242 50	1.287	1.77603	.563	34 9	850
900	533 66	514 03	644 71	85 91	730 62	1244 65	1 289	1 87804	.532	33 0	900
950	540 32	521 30	639 60	85 80	725 40	1246 70	1 291	1.98004	.505	31 4	950
1000	546 80	528 30	634 68	85 68	720 36	1248 66	1.293	2.08203	.480	30.0	1000

The column headed "*U*" in the table of the properties of saturated steam is useful for reducing the performance of different boilers to a common standard—this standard being that most generally accepted by engineers: the equivalent evaporation at atmospheric pressure and the temperature of boiling water, or, as it is frequently called, the evaporation from and at 212°. In the table it is assumed that the temperature of the feed-water is 32°, and an auxiliary table is added, giving corrections for any temperature of feed from 32° to 212°.

CORRECTION FOR TOTAL HEAT IN UNITS OF EVAPORATION.

Temperature of feed, Fahrenheit degrees.	Correction.	Temperature of feed, Fahrenheit degrees.	Correction.	Temperature of feed, Fahrenheit degrees.	Correction.	Temperature of feed, Fahrenheit degrees.	Correction.	Temperature of feed, Fahrenheit degrees.	Correction.
33	.0010	69	.0383	105	.0756	141	.1129	177	.1504
34	.0021	70	.0393	106	.0766	142	.1140	178	.1514
35	.0031	71	.0404	107	.0777	143	.1150	179	.1525
36	.0041	72	.0414	108	.0787	144	.1160	180	.1535
37	.0052	73	.0424	109	.0797	145	.1171	181	.1545
38	.0062	74	.0435	110	.0808	146	.1181	182	.1556
39	.0073	75	.0445	111	.0818	147	.1192	183	.1566
40	.0083	76	.0456	112	.0829	148	.1202	184	.1577
41	.0093	77	.0466	113	.0839	149	.1213	185	.1587
42	.0104	78	.0476	114	.0849	150	.1223	186	.1598
43	.0114	79	.0487	115	.0860	151	.1233	187	.1608
44	.0124	80	.0497	116	.0870	152	.1244	188	.1618
45	.0135	81	.0507	117	.0880	153	.1254	189	.1629
46	.0145	82	.0518	118	.0891	154	.1264	190	.1639
47	.0155	83	.0528	119	.0901	155	.1275	191	.1650
48	.0166	84	.0538	120	.0911	156	.1285	192	.1660
49	.0176	85	.0549	121	.0922	157	.1296	193	.1670
50	.0186	86	.0559	122	.0932	158	.1306	194	.1681
51	.0197	87	.0569	123	.0943	159	.1316	195	.1691
52	.0207	88	.0580	124	.0953	160	.1327	196	.1702
53	.0217	89	.0590	125	.0963	161	.1337	197	.1712
54	.0228	90	.0601	126	.0974	162	.1348	198	.1723
55	.0238	91	.0611	127	.0984	163	.1358	199	.1733
56	.0248	92	.0621	128	.0994	164	.1368	200	.1743
57	.0259	93	.0632	129	.1005	165	.1379	201	.1754
58	.0269	94	.0642	130	.1015	166	.1389	202	.1764
59	.0279	95	.0652	131	.1025	167	.1400	203	.1775
60	.0290	96	.0663	132	.1036	168	.1410	204	.1785
61	.0300	97	.0673	133	.1046	169	.1420	205	.1796
62	.0311	98	.0683	134	.1057	170	.1431	206	.1806
63	.0321	90	.0694	135	.1067	171	.1441	207	.1817
64	.0331	100	.0704	136	.1077	172	.1452	208	.1827
65	.0342	101	.0714	137	.1088	173	.1462	209	.1837
66	.0352	102	.0725	138	.1098	174	.1473	210	.1848
67	.0362	103	.0735	139	.1109	175	.1483	211	.1858
68	.0372	104	.0746	140	.1119	176	.1493	212	.1869

XI.

TOTAL AVAILABLE ENERGY IN WATER AND STEAM.

Pressure above a vacuum in pounds per square inch	Same pressure as indicated by steam-gauge, allowing 14.7 pounds for atmospheric pressure.	Absolute pressure in atmospheres.	Number of British thermal units required for the evaporation of one pound of water, known as latent heat of evaporation, *H*.	Temperature in degrees Fahrenheit of the steam and of the water from which it is evaporated.	Temperature in degrees Centigrade of the steam and of the water from which it is evaporated.	Corresponding absolute temperature in degrees Fahrenheit.	Corresponding absolute temperature in degrees Centigrade.	Amount of energy contained in one pound of water which may be liberated by explosion or expansion to 212° Fahr.	Corresponding amount of energy contained in the latent heat of evaporation.	Total amount of energy contained in one pound of steam at corresponding temperatures and pressures.
20	5.3	1.36	954.415	227.9	108.8	689.0	382.8	145.9	16872.9	17018.8
25	10.3	1.70	945.825	240.0	115.5	701.2	389.5	439.7	29136.8	29596.5
30	15.3	2.04	938.905	250.2	121.2	711.4	395.2	813.5	38921.9	39735.4
35	20.3	2.38	932.1523	259.1	126.1	720.3	400.1	1223.4	47054.9	48278.3
40	25.3	2.72	926.4728	267.1	130.1	728.3	404.6	1645.7	54111.7	55757.4
45	30.3	3.06	921.3343	274.2	134.5	735.4	408.5	2112.9	60158.1	62271.0
50	35.3	3.40	916.6316	280.8	138.2	742.0	412.2	2550.4	65613.8	68164.2
55	40.3	3.74	912.2906	286.8	141.5	748.0	415.5	2999.9	70428.7	73428.6
60	45.3	4.08	908.2479	292.5	144.7	753.7	418.7	3449.2	74884.6	78333.8
65	50.3	4.42	904.4621	297.7	147.6	758.9	421.6	3899.8	78850.5	82750.3
70	55.3	4.76	900.8991	302.7	150.4	763.9	424.4	4361.1	82577.7	86938.8
75	60.3	5.10	897.5269	307.3	152.9	768.5	426.9	4815.8	85923.6	90739.4
80	65.3	5.44	894.3304	311.8	155.4	773.0	429.4	5206.5	89138.7	94345.2
85	70.3	5.78	891.2862	316.0	157.7	777.2	431.7	5638.9	92073.3	97712.2
90	75.3	6.12	888.3758	320.0	160.0	781.2	434.0	6058.1	94814.7	100872.8
95	80.3	6.46	885.5887	323.8	162.1	785.0	436.1	6474.2	97447.2	103921.4
100	85.3	6.80	882.9144	327.5	164.1	788.7	438.1	6885.2	99787.6	106672.8
105	90.3	7.14	880.3429	331.1	166.1	792.3	440.1	7290.3	102163.3	109453.6
110	95.3	7.48	877.8653	334.5	168.0	795.7	442.0	7689.0	104334.9	112023.9
115	100.3	7.82	875.4721	337.8	169.8	799.0	443.8	8087.3	106421.7	114509.0
120	105.3	8.16	873.1555	340.9	171.6	802.1	445.6	8483.1	108325.4	116808.5
125	110.3	8.50	870.9115	344.0	173.3	805.2	447.3	8864.9	110219.9	119084.8
130	115.3	8.84	868.7351	347.0	175.0	808.2	449.0	9252.6	112025.6	121278.2
135	120.3	9.18	866.6223	349.9	176.6	811.1	450.6	9627.0	113745.7	123372.7
140	125.3	9.52	864.5061	352.7	178.1	813.9	452.1	9992.6	115382.1	125374.7

TOTAL AVAILABLE ENERGY IN WATER AND STEAM—*Continued.*

Pressure above a vacuum in pounds per square inch.	Same pressure as indicated by steam gauge, allowing 14.7 pounds for atmospheric pressure.	Absolute pressure in atmospheres.	Number of British thermal units required for the evaporation of one pound of water, known as latent heat of evaporation, *H*	Temperature in degrees Fahrenheit of the steam and of the water from which it is evaporated	Temperature in degrees Centigrade of the steam and of the water from which it is evaporated.	Corresponding absolute temperature in degrees Fahrenheit	Corresponding absolute temperature in degrees Centigrade.	Amount of energy contained in one pound of water which may be liberated by explosion or expansion to 212° Fahr.	Corresponding amount of energy contained in the latent heat of evaporation.	Total amount of energy contained in one pound of steam at corresponding temperatures and pressures.
145	130.3	9.86	862.5679	355.5	179.7	816.7	453.7	10361.0	117003.5	127364.5
150	135.3	10.20	860.6213	358.1	181.1	819.3	455.1	10536.5	118477.2	129003.7
155	140.3	10.54	858.7276	360.7	182.6	821.9	456.6	11085.9	119939.4	131025.3
160	145.3	10.88	856.8740	363.2	184.0	824.4	458.0	11444.2	121323.6	132767.8
165	150.3	11.22	855.0654	365.7	185.4	826.9	459.4	11823.4	122697.8	134521.2
170	155.3	11.56	853.2942	368.1	186.7	829.3	460.7	12141.3	123995.5	136136.8
175	160.3	11.90	851.5670	370.5	188.0	831.7	462.0	12408.7	125284.7	137793.4
180	165.3	12.24	849.8668	372.8	189.3	834.0	463.3	12831.4	126499.1	139320.5
185	170.3	12.58	848.2086	375.0	190.5	836.2	464.5	13182.0	127642.4	140824.4
190	175.3	12.92	846.5844	377.2	191.7	838.4	465.7	13367.1	128778.8	142145.9
195	180.3	13.26	844.9238	379.4	193.0	840.6	467.0	13844.1	129908.3	143757.4
200	185.3	13.60	843.4326	381.5	194.1	842.7	468.1	14153.3	130967.4	145120.7
210	195.3	14.28	840.3967	385.6	196.4	846.8	470.4	14830.8	133003.2	147834.0
220	205.3	14.66	838.5864	389.8	198.7	851.0	472.7	15463.1	136003.3	151466.4
230	215.3	15.64	833.9691	394.2	201.2	855.4	475.2	16180.3	137134.2	153314.7
240	225.3	16.32	832.6419	397.9	203.3	854.1	477.3	16790.2	139094.3	155884.5
250	235.3	17.00	830.3630	401.0	205.0	862.2	479.0	17314.4	140516.0	157830.4
500	485.3	34.01	780.8592	467.6	242.0	898.8	516.0	30055.5	165892.7	195948.2
1000	985.3	68.02	720.4350	546.8	286.0	1008.0	560.0	48671.5	179812.0	227883.5
2000	1985.3	136.05	643.9249	643.7	339.8	1104.9	613.8	75777.2	194221.3	269998.5
3000	2985.3	204.08	590.8038	708.3	375.7	1169.5	649.7	96116.3	193555.0	289671.3
4000	3985.3	272.10	544.6774	763.7	405.9	1223.9	679.9	114498.6	189201.7	301700.3
5000	4985.3	340.13	505.7339	809.8	431.0	1269.0	705.1	130494.2	183305.9	313800.1
6000	5985.3	408.16	471.8473	845.9	452.1	1307.1	726.1	144413.4	176656.9	321070.3
7000	6985.3	486.19	439.9985	881.2	471.7	1342.4	745.7	157944.2	169133.3	327277.5
8000	7985.3	544.21	409.4533	914.3	490.1	1375.5	764.1	170837.0	161192.6	332224.6
9000	8985.3	612.24	382.6347	945.2	507.3	1406.4	781.5	183998.6	153998.0	337142.6
10000	9985.3	680.27	355.2491	971.4	521.8	1437.6	795.8	193787.0	145376.8	339164.2
100000	99985.3	6802.72	305.3040	2163.4	1184.1	2624.6	1458.1			

XII.

FORMULAS RELATING TO PROPERTIES OF STEAM.

QUANTITY.			SYMBOL.	FORMULA.
Pressure.	Above a Vacuum.	Pounds per square inch.	P	$P = \frac{p}{144}$, $\log P = 6.1007 - \frac{2731.62}{t} - \frac{396944}{t^2}$
		Pounds per square foot.	p	$p = P \times 144$, $\log p = 8.2591 - \frac{2731.62}{T} - \frac{396944}{T^2}$
		Inches of mercury, at 32° Fahr.	M	$M = P \times 2.03759$
		Feet of distilled water, at temperature of maximum density.	F	$F = P \times 2.306768$
		Atmospheres.	A	$A = P \times 0.0680967$
	Above the atmosphere, in pounds per square inch.		G	$G = P - 14.685$
Temperature.	Fahrenheit's scales.		t	$t = T - 461^\circ.2$
	Absolute scale, Fahrenheit degrees.		T	$T = 1 \div \left(\sqrt{\frac{8.2591 - \log p}{396944} + 0.00001184} - 0.003441\right)$
Quantity of heat.	Per pound of steam in British thermal units.	Required to raise the temperature of the water from 32° to t°.	S	$S = t - 32 + 0.000000103(t - 39.1)^3$
		Required to change the water into steam. (Internal latent heat.)	I	$I = L - E$
		Required to overcome the pressure of the surrounding medium. (External latent heat.)	E	$E = p \times \frac{C - v}{772}$
		Latent heat of evaporation, under constant pressure, P.	L	$L = 1091.7 - 0.695(t - 32) - 0.000000103(t - 39.1)^3$
		Total heat of evaporation above 32°.	H	$H = 1091.7 + 0.305(t - 32)$
	Total heat of evaporation per pound of steam, above 32°, in units of evaporation.		U	$U = \frac{H}{966.1}$

FORMULAS RELATING TO PROPERTIES OF STEAM—*Continued.*

QUANTITY.		SYMBOL.	FORMULA.
Foot-pounds of energy, in latent heat of evaporation, per cubic foot of steam.		l	$l = 2.3026 \times p \times \left(\frac{2731.62}{T} + \frac{793888}{T^2}\right)$
Weight.	Of a cubic foot of steam, in pounds.	W	$W = \frac{l}{772 \times L}$
	Of a cubic foot of distilled water, in pounds, at temperature t.	w	$w = \frac{62.425}{v}$
Volume.	Of a pound of steam, in cubic feet.	C	$C = \frac{1}{W}$
	Ratio of volume of steam to volume of equal weight of distilled water at temperature of maximum density.	V	$V = C \times 62.425$
	Ratio of volume of distilled water, at temperature T, to volume of equal weight at temperature of maximum density.	v	For temperatures from 32° to 70°, $v = 1.00012 - 0.000033914(t-32) + 0.0000083822(t-32)^2 - 0.000000006403(t-32)^3$. For temperatures above 70°, $v = 0.99781 + 0.00006117(t-32) + 0.000001059(t-32)^2$

FACTORS OF EVAPORATION.

Temperature of Feed-water in Degrees. F.	C.	25 / 1.7	30 / 2.0	35 / 2.3	40 / 2.7	45 / 3.0	50 / 3.3	60 / 4.0	70 / 4.7	80 / 5.3	90 / 6.0	100 / 6.7	120 / 8.0	140 / 9.3	160 / 10.7	180 / 12	200 / 13.3
		Gauge Pressure in Pounds per Square Inch Above the Atmosphere and in Atmospheres.															
32	0	1.204	1.206	1.209	1.211	1.212	1.214	1.217	1.219	1.222	1.224	1.227	1.231	1.234	1.237	1.239	1.241
35	1.6	.201	.203	.206	.208	.209	.211	.214	.216	.219	.221	.224	.228	.231	.234	.236	.238
40	4.4	.196	.198	.201	.203	.204	.206	.209	.211	.214	.216	.219	.223	.226	.229	.231	.233
45	7.2	.190	.192	.195	.197	.198	.200	.203	.205	.208	.210	.213	.217	.220	.223	.225	.227
50	10	.185	.187	.190	.192	.193	.195	.198	.200	.203	.205	.208	.212	.215	.218	.220	.222
55	12.7	.180	.182	.185	.187	.188	.190	.193	.195	.198	.200	.203	.207	.210	.213	.215	.217
60	15.5	.175	.177	.180	.182	.183	.185	.188	.190	.193	.195	.198	.202	.205	.208	.210	.212
65	18.3	.170	.172	.175	.177	.178	.180	.183	.185	.188	.190	.193	.197	.200	.203	.205	.207
70	21.1	.165	.167	.170	.172	.173	.175	.178	.180	.183	.185	.188	.192	.195	.198	.200	.202
75	23.5	.160	.162	.165	.167	.168	.170	.173	.175	.178	.180	.183	.187	.190	.193	.195	.197
80	26.6	.154	.156	.159	.161	.162	.164	.167	.169	.172	.174	.177	.181	.184	.187	.189	.191
85	29.4	.149	.151	.154	.156	.157	.159	.162	.164	.167	.169	.172	.176	.179	.182	.184	.186
90	32.2	.144	.146	.149	.151	.152	.154	.157	.159	.162	.164	.167	.171	.174	.177	.179	.181
95	35.0	.139	.141	.144	.146	.147	.149	.152	.154	.157	.159	.162	.166	.169	.172	.174	.176
100	37.7	.134	.136	.139	.141	.142	.144	.147	.149	.152	.154	.157	.161	.164	.167	.169	.171
105	40.5	.128	.130	.133	.135	.136	.133	.141	.143	.146	.148	.151	.155	.158	.161	.163	.165
110	43.3	.123	.125	.128	.130	.131	.133	.136	.138	.141	.143	.146	.150	.153	.156	.158	.160
115	46.1	.118	.120	.123	.125	.126	.120	.131	.133	.136	.138	.141	.145	.143	.151	.153	.155
120	48.8	.113	.115	.118	120	.121	.123	.126	.128	.131	.133	.136	.140	.143	.146	.148	.150
125	51.6	.108	.110	.113	.115	.116	.118	.121	.123	.126	.128	.131	.135	.138	.141	.143	.145
130	54.4	.102	.104	.107	.109	.110	.112	.115	.117	.120	.122	.125	.129	.132	.135	.137	.139
135	57.2	.097	.099	.102	.104	.105	.107	.110	.112	.115	.117	.120	.124	.127	.130	.132	.134
140	60.0	.092	.094	.097	.099	.100	.102	.105	.107	.110	.112	.115	.119	.122	.125	.127	.129
145	62.7	.087	.089	.092	.094	.095	.097	.100	.102	.105	.107	.110	.114	.117	.120	.122	.124
150	65.5	.082	.084	.087	.089	.090	.092	.095	.097	.100	.102	.105	.109	.112	.115	.117	.119
155	68.3	.076	.078	.081	.083	.084	.086	.089	.091	.094	.096	.099	.103	.106	.109	.111	.113
160	71.1	.071	.073	.076	.078	.079	.081	.084	.086	.089	.091	.094	.098	.101	.104	.106	.108
165	73.8	.066	.068	.071	.073	.074	.076	.079	.081	.084	.086	.089	.093	.096	.099	.101	.103
170	76.6	.061	.063	.066	.068	.069	.071	.074	.076	.079	.081	.084	.088	.091	.094	.096	.098
175	79.4	.056	.058	.061	.063	.064	.066	.069	.071	.074	.076	.079	.083	.086	.089	.091	.093
180	82.2	.050	.052	.055	.057	.058	.060	.063	.065	.068	.070	.073	.077	.080	.083	.085	.087
185	85.0	.045	.047	.050	.052	.053	.055	.058	.060	.063	.065	.068	.072	.075	.078	.080	.082
190	87.7	.040	.042	.045	.047	.048	.050	.053	.055	.058	.060	.063	.067	.070	.073	.075	.077
195	90.5	.035	.037	.040	.042	.043	.045	.048	.050	.053	.055	.058	.062	.065	.067	.070	.072
200	93.3	.030	.032	.035	.037	.038	.040	.043	.045	.048	.050	.053	.057	.060	.063	.065	.067
205	96.1	.025	.027	.030	.032	.033	.035	.038	.040	.043	.045	.048	.052	.055	.058	.060	.062
210	98.8	.020	.022	.025	.027	.028	.030	.033	.035	.038	.040	.043	.047	.050	.053	.055	.057

XIV.

COMPOSITION OF VARIOUS FUELS OF THE UNITED STATES.

	C.	H.	O.	N.	S.	Moisture.	Ash.	Spec. Grav.
Pennsylvania Anthracite	78.6	2.5	1.7	0.8	0.4	1.2	14.8	1.45
Rhode Island "	85.8	10.5		....	3.7	....	....	1.85
Massachusetts "	92.0	6.0		....	2.0	....	....	1.78
North Carolina "	83.1	7.8		...	9.1	....	....	..
Welsh "	84.2	3.7	2.3	0.9	0.9	1.3	6.7	1.40
Maryland Semi-bituminous	80.5	4.5	2.7	1.1	1.2	1.7	8.3	1.33
Pennsylvania "	75.8	20.2		....	....	....	4.0	1.32
" "	59.4	38.8		....	....	....	1.8	1.30
Indiana "	70.0	28.0		....	..	....	2.0	1.24
" "	52.0	39.0		....	....	....	9.0	1.27
Illinois Bituminous	62.6	35.5		....	....	...	1.9	1.30
" (Block) Bituminous	58.2	37.1		....	...	....	4.7	....
Illinois and Indiana (Cannel) Bituminous	59.5	36.6		....	....	....	3.9	1.27
Kentucky (Cannel) Bituminous	48.4	48.8		....	....	....	2.8	1.25
Tennessee Bituminous	71.0	17.0		....	.. .	....	12.0	1.45
" "	41.5	56.5		....	.	.	2.5	.
Alabama "	54.0	42.6		....	1.0	1.2	1.2	....
Virginia "	55.0	41.0		...	....	....	4.0	....
" "	74.0	18.6		....	..	....	7.4	....
California and Oregon Lignite	50.1	3.9	13.7	0.9	1.5	16.7	13.2	1.32

Coal. State.	Coal. Kind of Coal.	Per Cent. of Ash.	Theoretical Value. In Heat Units.	Theoretical Value. In Pounds of Water Evaporated.
Pennsylvania	Anthracite	3.49	14.199	14.70
"	"	6.13	13.535	14.01
"	"	2.90	14.221	14.72
"	Cannel	15.02	13.143	13.60
"	Connelsville	6.50	13.368	13.84
"	Semi-bituminous	10.77	13.155	13.62
"	Stone's Gas	5.00	14.021	14.51
"	Youghiogheny.	5.60	14.265	14.76
"	Brown	9.50	12.324	12.75
Kentucky	Caking	2.75	14.391	14.89
"	Cannel	2.00	15.198	16.76
"	"	14.80	13.360	13.84
"	Lignite	7.00	9.326	9.65
Illinois	Bureau County	5.20	13.025	13.48
"	Mercer County	5.60	13.123	13.58
"	Montauk	5.50	12.659	13.10
Indiana	Block	2.50	13.588	14.38
"	Caking	5.66	14.146	14.64
"	Cannel	6.00	13.097	13.56
Maryland	Cumberland	13.98	12.226	12.65
Arkansas	Lignite	5.00	9.215	9.54
Colorado	"	9.25	13.562	14.04
"	"	4.50	13.866	14.35
Texas	"	4.50	12.962	13.41
Washington.	"	3.40	11.551	11.96
Pennsylvania	Petroleum	....	20.746	21.47

ANALYSES OF ASH.

	Specific Grav.	Color of Ash.	Silica.	Alumina.	Oxide Iron.	Lime.	Magnesia.	Loss.	Acids S.&P.
Pennsylvania Anthracite.....	1.559	Reddish Buff.	45.6	42.75	9.43	1.41	0.33	0.48	....
" Bituminous....	1.372	Gray.	76.0	21.00	2.60	....	...	0.40	....
Welsh Anthracite............	1.32	....	40.0	44.8	....	12.0	trace	....	2.97
Scotch Bituminous...........	1.26	...	37.6	52.0	....	3.7	1.1	....	5.02
Lignite........................	1.27	....	19.3	11.6	5.8	23.7	2.6	....	33.8

XV.

HORSE-POWER PER POUND MEAN PRESSURE.

Diameter of Cylinder. Inches.	Speed of Piston in Feet per Minute.										
	100	240	300	350	400	450	500	550	600	650	750
4	.038	.091	.114	.133	.152	.171	.19	.209	.228	.247	.285
4½	.048	.115	.144	.168	.192	.216	.24	.264	.288	.322	.360
5	.06	.144	.18	.21	.24	.27	.30	.33	.36	.39	.450
5½	.072	.173	.216	.252	.288	.324	.36	.396	.432	.468	.540
6	.086	.205	.256	.299	.342	.385	.428	.471	.513	.555	.641
6½	.102	.245	.307	.391	.409	.464	.512	.563	.614	.698	.800
7	.116	.279	.348	.408	.466	.524	.583	.641	.699	.756	.874
7½	.134	.321	.401	.468	.534	.602	.669	.735	.802	.869	1 009
8	.152	.365	.456	.532	.608	.685	.761	.837	.912	.989	1 141
8½	.172	.413	.516	.602	.688	.774	.86	.946	1 032	1 118	1 290
9	.192	.462	.577	.674	.770	.866	.963	1 059	1 154	1.251	1 444
9½	.215	.515	.644	.751	.859	.966	1 074	1 181	1 288	1 395	1 610
10	.238	.571	.714	.833	.952	1 071	1 190	1 309	1 428	1 547	1 785
10½	.262	.63	.787	.919	1 050	1 181	1 313	1 444	1 575	1 706	1 969
11	.288	.691	.804	1 008	1 152	1 296	1 44	1 584	1 728	1 872	2 160
11½	.314	.754	.943	1 1	1 257	1 414	1 572	1 729	1 886	2.043	2 357
12	.342	.820	1 025	1 195	1 366	1 540	1 708	1 880	2 050	2 222	2 564
13	.402	.964	1 206	1 407	1 608	1 809	2 01	2 211	2 412	2 613	3 015
14	.466	1 119	1 398	1 631	1 864	2 097	2 331	2 564	2 797	3 029	3 495
15	.535	1 285	1 606	1 873	2 131	2 409	2 677	2 945	3 212	3 479	4 004
16	.609	1 461	1 827	2 131	2 436	2 741	3 045	3 349	3 654	3 958	4 567
17	.685	1 643	2 054	2 396	2 739	3 081	3 424	3 766	4 108	4 450	5 135
18	.771	1 849	2 312	2 697	3 083	3 468	3 854	4 239	4 624	5 009	5 780
19	.859	2 061	2 577	3 006	3 436	3 865	4 295	4 724	5 154	5 583	6 442
20	.952	2 292	2 855	3 331	3 807	4 285	4 759	5 234	5 731	6 186	7 138
21	1 049	2 518	3.148	3 672	4 197	4 722	5 247	5 771	6 296	6 820	7 869
22	1 152	2 764	3 455	4 031	4 607	5 183	5 759	6 334	6 911	7 486	8 638
23	1 259	3 021	3 776	4 405	5 035	5 664	6 294	6 923	7 552	8 181	9 44
24	1 370	3 289	4 111	4 797	5 482	6 167	6 853	7 538	8 223	8 908	10 279
25	1 487	3 569	4 461	5 105	5 948	6 692	7 436	8 179	8 923	9 566	11 053
26	1 609	3 861	4 826	5 630	6 435	7 239	8 044	8 848	9 652	10 456	12 065
27	1 733	4 159	5 199	6.066	6 932	7 799	8 666	9 532	10 399	11 265	12 998
28	1 865	4 477	5 596	6 529	7 462	8 395	9 328	10 261	11 193	12 125	13 991
29	2 002	4 805	6 006	7 007	8 008	9 009	10.01	11 011	12 012	13 013	15 015
30	2 142	5 141	6 426	7 497	8 568	9 639	10.71	11 781	12 852	13 923	16 065
31	2 288	5 486	6 865	8 001	9 144	10 787	11 43	12 573	13 716	14 866	17 145
32	2 436	5 846	7 308	8 526	9 744	10 962	12.18	13 398	14 616	15 834	18 270
33	2 590	6 216	7 770	9 065	10 360	11 655	12 959	14 245	15 54	16 835	19 425
34	2 746	6 59	8 238	9 611	10 984	12 357	13 73	15 103	16 476	17 849	20 595
35	2 914	6 995	8 742	10 199	11 656	13 113	14 57	16 027	17 484	18 941	21 855
36	3 084	7 401	9 252	10 794	12 336	13 878	15 42	16 962	18 504	20 046	23 130
37	3 253	7 817	9 774	11 403	13 032	14 861	16 29	17 919	19 548	21 177	24 435
38	3 436	8 246	10 308	12 026	13 744	15 462	17 18	18 898	20 616	22 334	25 770
39	3 620	8 648	10 86	12 67	14 48	16 29	18 1	19 91	21 62	23 53	27 150
40	3 808	9 139	11 424	13 328	15 232	17 136	19 04	20.944	22 848	24 752	28 560
41	4 002	9 604	12 006	14 007	16 008	18 009	20 00	22 011	24 012	26 013	30 015
42	4 198	10 075	12 594	14 693	16 792	18 901	20 99	23 089	25 188	27 287	31 485
43	4 40	10 56	13 20	15 4	17 6	19 8	22 00	24 2	26 4	28 6	33 00
44	4 606	11 046	13 818	16 121	18 424	20 727	23 03	25 333	27 636	29 939	34 545
45	4 818	11 563	14 454	16 863	19 272	21 681	24 09	26 399	28 908	31 317	36 135
46	5 043	12 080	15 128	17 626	20 144	22 662	25 18	27 698	30 216	32 754	37 770
47	5 256	12 614	15 768	18 396	21 024	23 652	26 28	28 908	31 536	34 164	39 420
48	5 482	12 846	16 446	19 187	21 928	24 669	27 41	30 151	32 152	35 633	41 115
49	5 714	13 713	17 142	19 999	22 856	25 713	28 57	31 427	34 284	37 141	42 855
50	5 950	14 28	17 85	20 825	23 8	26 775	29 75	32 725	35 7	38 675	44 625
51	6 180	14 832	18 54	21 665	24 76	27 855	30 95	34 045	37 08	40 205	46 425
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57	7 732	18 557	23 196	27 062	30 928	34 794	38 66	42 526	46 392	50 258	57 99
58	8 006	19 214	24 018	28 021	32 024	36 027	40 03	44 033	48 036	52 039	60 045
59	8 284	19 902	24 852	28 964	33 136	37 278	41 42	45 562	48 704	53 846	62 13
60	8 506	20 558	25 698	29 981	34 264	38 547	42 83	47 113	51 396	55 679	64.245

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